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THEORETICAL RESEARCH OF GANTRY TRACTOR TURNING

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Abstract

Turning the gantry tractor on the turning lane can be done in different ways. An analysis of publications on the use of gantry tractors in the track system showed that the most common turn schemes involve pneumatic control. The theory of static and dynamic rotation of a wheeled vehicle has been sufficiently studied by scientists around the world. The aim of the research is to increase the efficiency of the process of turning the gantry tractor in the coordinate transport system of its operation, by studying its static and dynamic manoeuvrability. Theoretical research was carried out by modelling the conditions of the gantry tractor operation on the PC using the provisions of theoretical mechanics and tractor theory. Experimental studies of the gantry tractor were carried out according to both generally accepted and developed methods and involved the use of modern and specially designed equipment. Processing of experimental data was performed on a PC using regression and correlation-spectral analyses. As a result of the research, new schemes of turning such tractor are justified. It is proven that the speed of its rotation by this method should not exceed 1 $m \cdot s^{-1}$, and the ratio of its wheelbase should be minimal.

Keywords: gantry tractor; permanent technological track; coordinate transport system; manoeuvrability; theoretical and experimental research.

1. Introduction

Gantry devices for routed agriculture [1], [2], [3], [4], [6], [7], [8] should be equipped with a wheeled engine, which should be considered the most acceptable. The use of a crawler or other known type of propulsion in the presence of an improved track or road is impractical. [5] The theory of motion and controllability of wheeled traction and transport machines has been sufficiently developed by us [2], [3], [4]. The operation of the wheeled machine takes place under the action of a large number of perturbing factors (forces and their moments), which change the position of the machine in space and divert its movement from a given trajectory. The steering of modern wheeled machines is adapted to manual control and is built on the kinematic or power principle of turning.

When kinematic rotation, the schemes of turning steered wheels (front, rear or both front and rear) relative to the car body, or changing the position of one part of the mobile machine relative to another in the horizontal plane (articulated frame). Power (on-board) rotation is realized by rotating the wheels of different sides of the machine at different speeds. The most widespread schemes of turn with steered wheels. At the same time, if the machine has all the steered wheels, they can turn simultaneously on the front and rear axles in different directions (full-turn mode or “path way”) or to one side (crabbing mode) [13], [15]. Sometimes combined kinematic-force patterns are used or proposed for use, which improve the steerability of mobile machines [13], [15].

Traditional turning of wheeled machines is carried out on a certain area and includes several phases: entry into the turn, the actual turn, exit from the turn. Due to the presence of elasticities in the system “propulsion – support surface” the trajectory of the mobile machine when turning does not have a clear dependence on the rotation of the steering wheel, which complicates the construction of machines for turning [9], [10], [11].

Some scientists believe that effective in the automatic driving of vehicles within the coordinate transport system may be a way to change the direction of movement by turning the stopped car on the spot, for example, in a balanced state [12]. To implement this method of changing the direction of movement, the wheeled vehicle must be equipped with lifting and turning mechanisms, software device, position sensors relative to the intersection and control system. A machine turning pattern with all wheels steered has advantages over a weighted turning pattern: There is no need for a special lifting mechanism. But for automated machines with purely on-board turning and rigid mounting of all wheels, the preference must be given to a weighted headland. Therefore, the issue of modelling the static and dynamic manoeuvrability of the gantry tractor in the coordinate transport system of its operation is relevant and an enhanced scheme for turning such machinery is necessary. The purpose of research is to increase the efficiency of the process of turning the gantry tractor in the coordinate transport system of its operation, by studying its static and dynamic manoeuvrability.

2. Materials and methods

Theoretical research was carried out by modelling on a PC the conditions of operation of the gantry tractor using the provisions of theoretical mechanics and tractor theory. Experimental studies of the gantry tractor were carried out according to both generally accepted and developed methods and involved the use of modern and specially designed equipment. Processing of experimental data was performed on a PC using regression and correlation-spectral analyses.

Experimental studies were conducted in the greenhouse complex, which was equipped with a specially equipped laboratory for testing a gantry tractor. Turning of the gantry tractor was carried out by means of basic racks of power jacks (Fig. 1) placed on the bottom surface of its platforms for raising of the right or left boards accordingly. The time to perform the manoeuvre of turning (t_{tur}) of the gantry tractor in this way was determined by the sum of the duration of the individual actions:

$$t_{tur} = t_1 + t_2 + t_3 + t_4 + t_5 \quad (1)$$

where t_1 – the duration of the rise of the working bodies in the transport position, s;
t_2 – the duration of lifting one of the sides of the gantry tractor by extending the rod of the power jack, s;
t_3 – the duration of rotation of the gantry tractor around the support strut of the power jack, s;
t_4 – duration of lowering the board of the gantry tractor on the supporting surface of the traces of the constant technological track by the return stroke of the rod of the power jack, s;
t_5 – the duration of the transfer of working bodies from the transport position to the working one, s.

Fig. 1. Technical implementation of turning a gantry tractor

The angular velocity of rotation of the drive wheel was found in this way:

$$\omega_k = \frac{\pi \cdot n_k}{30} \quad (2)$$

n_k – the number of revolutions of the wheel of the gantry tractor (s^{-1}), which was defined as follows:

$$n_k = \frac{N_{ik} - 1}{2t_{ik}} \tag{3}$$

where N_{ik} – the total number of pulses received for analysis received from the speed sensor of the drive wheel of the gantry tractor during t_{ik}. Taking into account (3) the calculated working speed of the gantry tractor (V_{0p}, $m \cdot s^{-1}$):

$$V_{0p} = \omega_k \cdot R_k = \frac{L_k \cdot (N_{ik} - 1)}{120 \cdot t_{ik} \cdot (N - 1)} \tag{4}$$

Comparing the actual operating speed V_0 of the gantry tractor on the test section of the experimental laboratory with the theoretical V_{0p} estimated measurement error of this parameter:

$$\varsigma_V = \frac{V_0 - V_{0p}}{V_0} \cdot 100 \tag{5}$$

Experimentally, the slip coefficient δ of the engines of the gantry tractor was determined by expression:

$$\delta = 1 - \frac{n_{kx}}{n_{k0}} \cdot \frac{V_0}{V_x} \tag{6}$$

where n_{kx} and n_{k0} – the number of revolutions of the wheel of the gantry tractor on the same section of the road at idle and operating speeds, respectively;
V_x and V_0 – the speed of the gantry tractor at idle and operating strokes, respectively.

In the process of experimental research, the tangential thrust force P_k, developed by the wheel of the gantry tractor was analytically evaluated by the sum of two forces realized by it – traction and rolling resistance:

$$P_k = P_h + P_f \tag{7}$$

where P_h – traction force, which develops the wheel of the gantry tractor;
P_f – rolling resistance overcoming the wheel of the gantry tractor.

3. Results and Discussion

To conduct research, consider the scheme of rotation of the gantry tractor (Fig. 4), according to which its rotation is carried out by turning the chassis platform with steered wheels from one side around the centre, located in the centre of the wheelbase on the other side. According to this scheme of turning of the gantry tractor its wheels from one board can remain within the limits of the transport and technological track. Technically it can be implemented by our proposed second scheme of turning the gantry tractor (see Fig. 4) in at least two ways:
1) kinematic rotation of the platform by steered wheels on one side by means of a hinged rotary mechanism located in the centre of the wheelbase on the other side;
2) kinematic-power reversal of the gantry tractor by hanging one of its sides by means of lifting mechanisms.

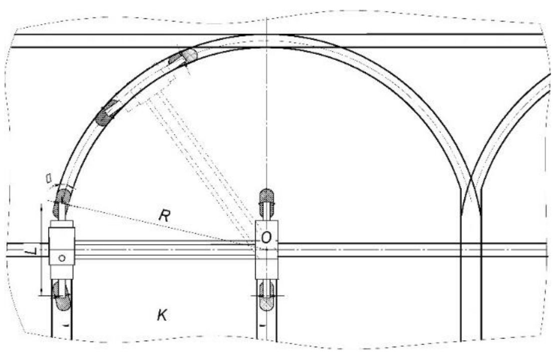

Fig. 2. The scheme of turning the gantry tractor around the centre, located in the centre of the wheelbase of one of its sides: *O* – centre of rotation; *K* – track width of the tractor; *R* – turning radius; *L* – wheelbase; α – the angle of rotation of the steered wheels

Complete information on the possibility of movement with specified kinematic parameters can be obtained by determining the power parameters (tangential thrust on the drive wheels, energy consumption of the gantry tractor to overcome the total torque, etc.), which characterize its static and dynamic manoeuvrability. To obtain the characteristics of static rotation, consider the circular motion of a flat model of a gantry tractor around the centre of rotation (point O) (Fig. 2) with a constant angular velocity $\omega = \text{const}$, $\dot{\omega} = 0$. The steered wheels of one of its sides (in this case – the left) are turned to the corners α_1 and β_1.

When compiling the equations of motion, it is based on the generally accepted notion of the correctness of the turn [8]. According to which – all the driving wheels of the tractor (in Fig. 2 – are the two wheels of the left side) move without sliding, and the intersection of their axes occurs at a point that is the centre of curvature of the path. The lateral input of the wheels is not taken into account, because the gantry tractor performs angular rotation around a fixed axis and its movement in the perpendicular direction is absent. It is considered only those elements of the gantry tractor which carry out plane parallel movement. Tangential forces of inertia and inertial moments of resistance to rotation of the gantry tractor and its technological part due to their small size are not taken into account in the study.

The moving coordinate system *xSy* is rigidly connected to the gantry tractor (see Fig. 2), the beginning of which will be placed in the centre of mass of its port side (point S). The x-axis is directed parallel to the longitudinal axis of the axle tractor, and its positive direction will be in the direction of movement.

In the turning mode, a number of appropriate forces act on the axle tractor (see Fig. 2). In order to carry out further actions, it is advisable to place the mass m_b divided into two parts (m_1 and m_2), falling on its left side exactly at these points. The impact of the resistance of the working elements of the technological part of the axle tractor is represented by the main vector F_{tr}, applied in the centre of resistance (point C), the direction of the action of which at the turn makes an angle γ with the longitudinal axis of the axle tractor. Reactions arising in the articulated pivot of an axle tractor can be represented by the forces F_{jx} and F_{jy}, applied at the points J, and the moment of resistance M_j.

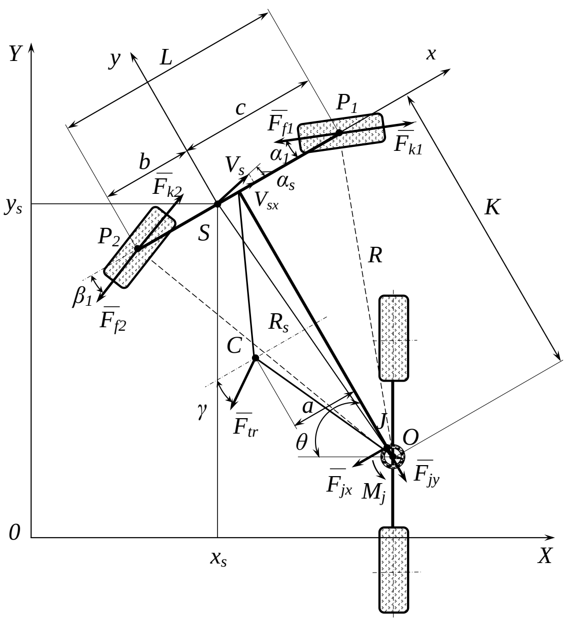

Fig. 3. Equivalent turning scheme of a (right-hand) gantry tractor with steered wheels on the left side

The equations of motion of an overhead tractor, describing the static rotation with respect to the moving coordinate system xSy, can be represented by the following system of known kinetostatic dependencies:

$$\left.\begin{aligned} m_b \cdot \mathrm{a}_{Sx} &= \sum F_{xj}\,, \\ m_b \cdot \mathrm{a}_{Sy} &= \sum F_{yj}\,, \\ \mathrm{M}_R &= \sum \mathrm{M}_{Si}\,, \end{aligned}\right\} \quad (8)$$

where a_{Sx}, a_{Sy} – the acceleration of the centre of mass (which is located at the point S) along the axes x and y the moving coordinate system;
$\sum F_{xj}$, $\sum F_{yj}$ – the sum of the projections of the j-th forces on the axes x and y the moving coordinate system;
M_R – total torque resistance;
$\sum \mathrm{M}_{Si}$ – is the sum of the i-th moments with respect to the centre of mass (point S).

In expanded form, system (8) will have expression:

$$\left.\begin{aligned}
&m_b \cdot \mathrm{a}_{Sx} = \left(F_{k1} - F_{f1}\right) cos\,\alpha_1 + \left(F_{k2} - F_{f2}\right) cos\,\beta_1 - F_{tr}\,cos\,\gamma - F_{j\mathrm{x}}\,cos\,\alpha_S + F_{jy}\,sin\,\alpha_S\,,\\
&m_b \cdot \mathrm{a}_{Sy} = -\left(F_{k1} - F_{f1}\right) sin\,\alpha_1 + \left(F_{k2} - F_{f2}\right) sin\,\beta_1 - F_{tr}\,sin\,\gamma - F_{jy}\,cos\,\alpha_S - F_{j\mathrm{x}}\,sin\,\alpha_S\,,\\
&M_R = c\left(F_{k1} - F_{f1}\right) sin\,\alpha_1 + b\left(F_{k2} - F_{f2}\right) sin\,\beta_1 - F_{tr}\,sin\,\gamma\,[0.5L + \mathrm{a}] + 0.5 \cdot K \cdot F_{tr}\,cos\,\gamma - M_j,
\end{aligned}\right\} \tag{9}$$

where α_S, a, b and c – constructive parameters of the gantry tractor, the nature of which is clear from Fig. 3.

To study the dynamic steering of an overhead tractor, the computational scheme (Fig 5.) is used. The tractor motion will be considered in a stationary coordinate system $X0Y$, connected with the plane of the turnstile. This allows to describe its motion by three independent generalized coordinates x_S, y_S, θ, where x_S, y_S – is the coordinates of the centre of mass (point S) when it moves in the stationary coordinate system $X0Y$; θ – is the angle between the abscissa axes of the moving and stationary coordinate systems.

The equations of plane-parallel motion of the gantry tractor, describing its dynamic rotation, with respect to the stationary coordinate system $X0Y$ represent the known dependencies:

$$\left.\begin{aligned}
&m_b \cdot \ddot{\mathrm{x}}_S = \textstyle\sum F_{\mathrm{X}j}\,,\\
&m_b \cdot \ddot{\mathrm{y}}_S = \textstyle\sum F_{\mathrm{Y}j}\,,\\
&J_{zS} \cdot \ddot{\theta} = \textstyle\sum \mathrm{M}_i\,,
\end{aligned}\right\} \tag{10}$$

where $\ddot{\mathrm{x}}_S$ – acceleration of the centre of mass (point S), directed along the axis X of the stationary coordinate system;
$\ddot{\mathrm{y}}_S$ – acceleration of the centre of mass (point S), directed along the axis Y of the stationary coordinate system;
J_{zS} – moment of inertia relative to the vertical axis z, passing through the point S;
$\ddot{\theta}$ – angular acceleration of the gantry tractor.

To compile the dynamic equations of motion of the gantry tractor, the principle of D'Alembert is used:

$$\left.\begin{aligned}
&m_b \cdot \ddot{\mathrm{x}}_S = \left(F_{k1} - F_{f1}\right) \cdot cos(\theta + \alpha_1) + \left(F_{k2} - F_{f2}\right) \cdot cos(\theta - \beta_1) - F_{tr} \cdot cos(\theta - \gamma)\,,\\
&m_b \cdot \ddot{\mathrm{y}}_S = -\left(F_{k1} - F_{f1}\right) \cdot sin(\theta + \alpha_1) + \left(F_{k2} - F_{f2}\right) \cdot sin(\theta - \beta_1) - F_{tr} \cdot sin(\theta - \gamma)\,,\\
&J_{zS} \cdot \ddot{\theta} = c \cdot \left(F_{k1} - F_{f1}\right) \cdot sin\,\alpha_1 + b \cdot \left(F_{k2} - F_{f2}\right) \cdot sin\,\beta_1 - F_{tr} \cdot sin\,\gamma \cdot [0.5L + \mathrm{a}] - M_R.
\end{aligned}\right\} \tag{11}$$

Initial conditions (t = 0): $x_S = x_0$; $y_S = y_0$; $\theta = 0$; $\dot{x}_S = 0$; $\dot{y}_S = 0$; $\dot{\theta} = 0$.

From fig. 3, the correlation of kinematic radii R_S and R, translational V_{Sx} and circumferential velocity V_S of the centre of mass (point S) with the construction parameters of the overhead tractor is expressed:

$$R_S = \sqrt{(\mathrm{c} - 0.5 \cdot L)^2 + \mathrm{K}^2} = \sqrt{R^2 - c \cdot L + c^2} = \sqrt{K^2\left(1 + \frac{\mu^2}{4}\right) - c(\mu \cdot K + c)} \tag{12}$$

With the help of equations (11) and (12) influence of structural and kinematic parameters of the gantry tractor on the criteria of its dynamic rotation can be estimated, which will justify the optimal values of structural and force parameters when performing circular motion on a turn.

The driving forces F_{k1} and F_{k2} of the front and rear left wheels of the overhead tractor can be represented with sufficient accuracy for practice by the known dependence:

$$F_{k1} + F_{k2} \;=\; F_{f1} + F_{f2} + F_{tr} \tag{13}$$

The rolling resistance forces of the front and rear (left) wheels of the axle tractor F_{f1} and F_{f2} are obtained from the following well-known expression:

$$F_{f1} = \; f + m_1 \cdot g \tag{14}$$

$$F_{f2} = \; f + m_2 \cdot g \tag{15}$$

where f – rolling resistance coefficient;
g – free fall acceleration, m·s^{-2}.

Assuming that $F_{k1} = F_{k2}$ then, if $F_{f1} \neq F_{f2}$, the following dependencies are valid:

$$F_{k1} = F_{f1} + F_{tr} \cdot \left(\frac{F_{f1}}{F_{f1} + F_{f2}}\right) \tag{16}$$

$$F_{k1} = F_{f1} + F_{tr} \cdot \left(\frac{F_{f1}}{F_{f1} + F_{f2}}\right) \tag{17}$$

The mutual influence of the platform of the axle tractor chassis and the wheel bogie at the hinged mount (point *O*), in accordance with the generally accepted principle of replacing the rejected bonds by reactions, the forces F_{jx} and F_{jy}, as well as the moment of resistance M_j are represented. The following equations are valid for the above parameters:

$$F_j = \sqrt{F_{\mathrm{jx}}^2 + F_{jy}^2} \tag{18}$$

$$M_j = F_j \cdot f_j \cdot r_j \tag{19}$$

where F_j – the resulting reaction from the action of forces F_{jx} and F_{jy};
f_j – the coefficient of resistance of the rotating elements of the hinge;
r_j – pivot radius.

The projections of the circular velocity vector V_S of the left side centre of mass on the axes of the stationary coordinate system are:

$$V_{Sx} = V_c \cdot cos\, \alpha_S \tag{20}$$

$$V_{Sy} = V_c \cdot sin\, \alpha_S \tag{21}$$

where V_{Sx} – translational speed of the centre of mass of the board of the gantry tractor;
V_{Sy} – the speed of lateral displacement of the centre of mass of the board of the gantry tractor, $V_{Sy} = 0$;
α_S – angle between vectors of translational V_{Sx} and circular V_S velocity of centre of mass of the axle tractor.

In accordance with Fig. 3:

$$tan\, \alpha_S = \frac{\mathrm{c} - 0.5 \cdot L}{\mathrm{K}} \tag{22}$$

The projections of the velocity components V_S on the axes of the stationary coordinate system will be:

$$V_{S\mathrm{X}} = V_{Sx} \cdot cos\, \theta \tag{23}$$

$$V_{S\mathrm{Y}} = V_{S\mathrm{x}} \cdot sin\, \theta \tag{24}$$

By differentiating in time the velocities V_{Sx} and V_{Sy}, the acceleration of the centre of mass of the left side of the gantry tractor in the stationary coordinate system is obtained:

$$a_{S\mathrm{X}} = \dot{V}_{S\mathrm{x}} \cdot cos\, \theta - V_{S\mathrm{x}} \cdot \omega \cdot sin\, \theta \tag{25}$$

$$a_{S\mathrm{Y}} = \dot{V}_{S\mathrm{x}} \cdot sin\, \theta + V_{S\mathrm{x}} \cdot \omega \cdot cos\, \theta \tag{26}$$

where $\dot{V}_{S\mathrm{x}}$ – linear acceleration during translational movement of the tractor;
$\omega = \dot{\theta} = V_{S\mathrm{x}} \cdot R_S^{-2}$ – angular speed of rotation of the gantry tractor;
R_S – turning radius of the gantry tractor.

To determine the accelerations of the left-side centre of mass in the moving coordinate system, projection of the acceleration vectors a_{Sx} and a_{Sy} on the moving x and y axes:

$$a_{S\mathrm{x}} = a_{S\mathrm{X}} \cdot cos\, \theta + a_{S\mathrm{Y}} \cdot sin\, \theta \tag{27}$$

$$a_{S\mathrm{y}} = -a_{S\mathrm{X}} \cdot sin\, \theta + a_{S\mathrm{Y}} \cdot cos\, \theta \tag{28}$$

Taking into account relations (25, 26) after the transformation it is obtained:

$$a_{S\mathrm{x}} = \dot{V}_{S\mathrm{x}} \tag{29}$$

$$a_{Sy} = V_{Sx} \cdot \omega \tag{30}$$

Consider the motion of the centre of the front wheel of the left side of the overhead tractor (point P_1) (Fig. 3). Denoting $V_{1x} = \dot{x}_{p1}$ and $V_{1y} = \dot{y}_{p1}$. The coordinates of point P_1 through the coordinates of the centre of mass of the left side of the overhead tractor are expressed:

$$x_{p1} = x_S + c \cdot cos\,\theta \tag{31}$$

$$y_{p1} = y_S + c \cdot sin\,\theta \tag{32}$$

Differentiating equation (31, 32) in time, the speed of the front wheel of the left side is obtained, expressed because of the speed of the centre of mass of the left side in projections on the stationary coordinate axes:

$$\dot{x}_{p1} = \dot{x}_S - c \cdot \dot{\theta} \cdot sin\,\theta \tag{33}$$

$$y_{p2} = \dot{y}_S + c \cdot \dot{\theta} \cdot cos\,\theta \tag{34}$$

Similarly, the coordinates and speed of the centre of the rear wheel of the left side of the overhead tractor (point P_2) (Fig. 3) are obtained through the coordinates and speed of the centre of the left side of the overhead tractor:

$$x_{p2} = x_c - b \cdot cos\,\theta \tag{35}$$

$$y_{p1} = y_c - b \cdot sin\,\theta \tag{36}$$

$$\dot{x}_{p2} = \dot{x}_c + b \cdot \dot{\theta} \cdot sin\,\theta \tag{37}$$

$$y_{p2} = \dot{y}_c - b \cdot \dot{\theta} \cdot cos\,\theta \tag{38}$$

To solve the static rotation problem, it is assumed that the translational velocity of the gantry tractor board is constant $V_{Sx} = const$. Then linear accelerations are equal to $a_{Sx} = 0$ and $a_{Sy} = (V_{Sx})^2 \cdot (R_S)^{-1}$. Thus, in the system of three equations (9) the unknowns are three parameters: reactions F_{jx} and F_{jy}, as well as the moment of resistance to rotation M_j, which allows to estimate the static rotationality of the axle tractor. To determine the moment of inertia J_z the well-known Huygens-Steiner theorem about moments of inertia about parallel axes is used:

$$J_{Zs} = J_S + m \cdot \left(\frac{K}{2}\right)^2 \tag{39}$$

where J_S – the moment of inertia of the gantry tractor relative to its centre of mass.

If the gantry tractor in the longitudinal-horizontal plane can be represented as a projection of a rectangle with sides K and L, the moment of inertia about the vertical axis passing through its geometric centre (assuming that this point is the centre of mass) can be calculated with sufficient accuracy by:

$$J_S = \frac{1}{12} m \cdot (K^2 + L^2) \tag{40}$$

Taking into account (40), the moment of inertia J_z can be determined from the dependence:

$$J_{Zs} = \frac{1}{12} m \cdot (K^2 + L^2) + m \cdot \left(\frac{K}{2}\right)^2 \tag{41}$$

Three equations of the initial system (9) contain the following unknown parameters: acceleration $\ddot{x}_S$, directed along the axis X; acceleration $\ddot{y}_c$ – along the axis Y; angular acceleration $\ddot{\theta}$, traction forces of driving wheels F_{k1} and F_{k2}. Thus, the system of output dynamic equations (9) is uncertain. Therefore, a particular case according to the following considerations is necessary. Turning the gantry tractor on the headland contains two characteristic phases: acceleration to a certain speed V_S and circular motion with constant forward and angular velocity ω (Fig. 4). It is clear that the smaller is the angle θ, the smaller is the distance to be covered by the gantry tractor on the turn with the desired acceleration.

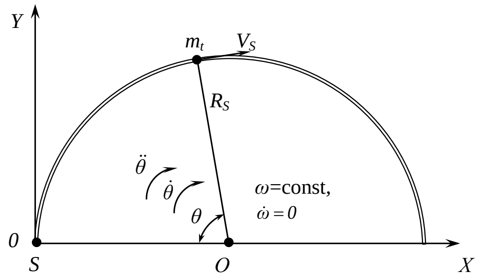

Fig. 4. Scheme explaining the mode phases of motion of the gantry tractor when turning in the fixed coordinate system *X0Y*

Under the initial conditions ($t = 0$) the origin of coordinates of the stationary system *X0Y* (see Fig. 4) is connected with the centre of the mass side of the gantry tractor (point *S*). In this case, at the initial moment of time:
$x_s = 0;\ y_s = 0;\ \theta_0 = 0;\ \dot{x}_s = 0;\ \dot{y}_s = 0.$
At the end of the 1st phase of turning the gantry tractor ($t = t_1$):
$x_s = x_{s1};\ y_s = y_{s1};\ \dot{x}_s = \dot{x}_{s1(max)};\ \dot{y}_s = \dot{y}_{s1(max)};\ \theta = \theta_1.$

The acceleration $\ddot{x}_s$ and $\ddot{y}_s$ and in the system of equations (9) is represented by derivatives of velocity $\dot{V}_{SX}$ and $\dot{V}_{Sy}$, $\ddot{\theta}$ – by $\dot{\omega}$. Then, it is taken into account that the trajectory of the centre of mass of the left side of the axle tractor is given by the angle θ, then variable t is expressed through the variable θ:

$$\ddot{x}_S = \frac{dV_{SX}}{dt} = \frac{dV_{SX}\cdot d\theta}{d\theta\cdot dt} = \frac{dV_{SX}}{d\theta}\cdot\omega = \frac{dV_{SX}}{d\theta}\cdot\frac{V_S}{R_S} \quad (42)$$

$$\ddot{y}_S = \frac{dV_{Sy}}{dt} = \frac{dV_{Sy}\cdot d\theta}{d\theta\cdot dt} = \frac{dV_{Sy}}{d\theta}\cdot\omega = \frac{dV_{Sy}}{d\theta}\cdot\frac{V_S}{R_S} \quad (43)$$

$$\ddot{\beta} = \frac{d\omega}{dt} = \frac{d\omega\cdot d\theta}{d\theta\cdot dt} = \frac{d\omega}{d\beta}\cdot\omega = \frac{d\omega}{d\theta}\cdot\frac{V_S}{R_S} \quad (44)$$

After integrating the equations of system (9) it is obtained:

$$V_{SX}{}^2 = \frac{R_S}{2\mathrm{M}}\cdot\{(F_{k1}-F_{f1})\cdot[sin(2\theta+\alpha_1)+2\theta\cdot cos\,\alpha_1+sin\,\alpha_1] - (F_{k2}-F_{f2})[sin(\beta_1-2\theta) - 2\theta\cdot cos\,\beta_1 - sin\,\beta_1] - F_{tr}[sin(2\theta+\gamma)+2\theta\cdot cos\,\gamma+sin\,\gamma]\} \quad (45)$$

$$V_{Sy}{}^2 = \frac{R_{\text{п}}}{2\mathrm{M}}\cdot\begin{cases}(F_{k1}-F_{f1})\cdot[sin(2\theta-\alpha_1)+2\theta\cdot cos\,\alpha_1+sin\,\alpha_1] - (F_{k2}-F_{f2})\cdot\\ [sin(\beta_1-2\beta)+2\beta\cdot cos\,\beta_1-sin\,\beta_1] - F_{tr}[sin(2\theta+\gamma)-2\theta\cdot cos\,\gamma+sin\,\gamma]\}\end{cases} \quad (46)$$

$$(V_{SX}{}^2+V_{Sy}{}^2) = \frac{2R_S{}^2}{J_z}\cdot\theta\{c\cdot(F_{k1}-F_{f1})\cdot sin\,\alpha_1 + b\cdot(F_{k2}-F_{f2})\cdot sin\,\alpha_2 - F_{\text{кр}}\cdot sin\,\gamma \cdot(0.5L+\mathrm{a}) - M_j \quad (47)$$

Specific traction force p_t, which is developed by an axle tractor, necessary to make a turn, it is defined as the ratio of tangential traction force F_k to its operating weight G_a [7]:

$$\mathrm{P}_t = \frac{F_k}{G_\mathrm{a}} \quad (48)$$

The physical essence of the p_t parameter is that as it decreases, the power inputs consumed by the axle tractor during curving on the headland decrease. The solution of the system (45,46,47) allowed us to establish the regularities of the impact of the speed mode of the gantry tractor on the specific traction force p_t, needed to make a turn (Table 1).

Preliminary calculations have established that the studied gantry tractor develops the maximum specific traction force $p_t = 0.05$. According to the data of Table 1, it is clear that when the latter makes a turn at low speed (0.5 $m\cdot s^{-1}$) the driving traction force realized by it allows to have the turning angle of the gantry tractor in the phase of its acceleration even less than 10 deg. With an increase in speed up to 1.0 $m\cdot s^{-1}$ the angle of rotation of the gantry tractor of the first phase of its acceleration is increased to 30 deg. Achieving the speed mode on the turn to 2.0 $m\cdot s^{-1}$ already becomes problematic. Because it is necessary to increase the acceleration phase of the specified axle tractor to the turning angle θ at least 140 deg. This means that almost all the way on the turn of the gantry tractor must move with acceleration, which is not desirable.

Specific traction force p_t, developed by an overhead tractor, is necessary to achieve the speed V_{Sx} on a turn			**Angle θ (deg) of rotation of the overhead tractor, accelerated to speed V_{Sx}**
V_{Sx} = 0.5 m·s^{-1}	V_{Sx} = 1.0 m·s^{-1}	V_{Sx} = 2.0 m·s^{-1}	
3.45E-02	1.35E-01	5.35E-01	10
1.79E-02	6.84E-02	2.68E-01	20
1.23E-02	4.62E-02	1.79E-01	30
9.55E-03	3.51E-02	1.35E-01	40
7.90E-03	2.85E-02	1.08E-01	50
6.75E-03	2.40E-02	9.06E-02	60
4.55E-03	1.52E-02	5.51E-02	100
3.60E-03	1.14E-02	3.99E-02	140
3.30E-03	1.02E-02	3.51E-02	160
3.10E-03	9.25E-03	3.14E-02	180

Table 1. Rotation characteristics of an overhead tractor according to the scheme in Fig. 4

Studies have shown that to obtain the optimal mode of rotation of the gantry tractor with increasing magnitude of their track, the value of the kinematic index of the mode of rotation also increases. In absolute value, the index of the turning mode varies from 5.8 m·rad^{-1} and reaches more than 30 m·rad^{-1} in the range of values of the track width of the gantry tractor 3-12 m.

The result of determining the specific traction force, which is necessary to perform a turn of an overhead tractor, according to the results of the study of dynamic turning, showed that when the latter makes a turn at low speed (0.5 m·s^{-1}) the driving traction force it realizes allows it to have an angle of turn in the phase of its acceleration even less than 10 deg, which is desirable. When the speed is increased to 1.0 m·s^{-1} the turning angle of the axle tractor's acceleration phase increases to 30 deg. Reaching the turning speed by 2.0 m·s^{-1} already becomes problematic. Because it is required to increase the acceleration phase of the specified axle tractor to a turning angle of not less than 140 deg. And this means that almost all the way on the turn the gantry tractor must move with acceleration, which is not desirable.

Turning an overhead tractor, the index of the turning mode significantly depends on the ratio of its wheelbase to the track width. When implementing the turning scheme by running all the steered wheels of the tractor around the centre of rotation, located in the centre of symmetry of its chassis, the index of the turning mode in the range of values of the track width of 3...12 m is in the range 2.3...20.0 m·rad^{-1}. When he realizes the rotation around the centre, located inside the interwheel space, this parameter varies from 5.8 m·rad^{-1} and reaches more than 30.0 m·rad^{-1}. Experimental timing for performing the power (onboard) turn manoeuvre (t_{tur}) of the axle tractor on the turning lane showed (Table 2) that the total duration of this process is about 1 min (0.0168 h).

Name of elementary time	**Duration, hours**
1. Lifting the working bodies in the transport position	0.0027
2. Lifting one of the sides of the tractor	0.0033
3. Rotate the tractor around the support jack of the power jack	0.0061
4. Lowering the tractor board	0.0025
5. Transfer of working bodies from the transport position to the working one	0.0022
Total time	0.0168

Table 2. Duration of the manoeuvre of power on-board rotation of the gantry tractor

This result is significant improvement in comparison with the duration of the turn manoeuvre of traditional large-capacity machine-tractor units. If it is possible to carry out this process in automatic mode, then the resulting duration of time to perform the turn manoeuvre with an overhead tractor can be reduced by at least half. The turning calculation and the new scheme can be used in automated movement and AI based applications, reducing the necessary time on non-effective movements.

4. Conclusion

The new scheme of turning the axle tractor on the headland by turning its chassis with steered wheels from one side around the centre located in the centre of the interwheel space from the other side, proposed by us, allows moving it simultaneously with turning to the next working position with better kinematic parameters. In practice, the best turning performance is achieved with an axle tractor construction when the ratio of its wheelbase to track width is less. The steering angle of its steering wheels is also the smallest under this condition.

The conducted researches show that in order to obtain the optimum mode of turning of an axle tractor with the increase of their path from 3 m to 12 m the kinematic index of the turning mode also increases from 5.8 $m \cdot rad^{-1}$ to 30 $m \cdot rad^{-1}$ and more. Practically, this requires an increase in the speed mode of their turning movement, which requires an increase in energy consumption for this process. The manoeuvre of the gantry tractor turn at low speed (0.5 $m \cdot s^{-1}$) allows you to have its turn angle in the acceleration phase even less than 10 deg. But when increasing the speed up to 1.0 $m \cdot s^{-1}$ the turning angle of the gantry tractor in the acceleration phase increases to 30 deg. Realization of the speed mode by an overhead tractor in a turn with speeds about 2.0 $m \cdot s^{-1}$ already becomes problematic. Because it is required to increase the phase of its acceleration to the turning angle not less than 140 deg. The new scheme could and will be used as an input to AI based and automated tractors.

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