Dynamic Calculation of a Steering Control of Gantry High-Clearance Tractor Used in Horticulture and Viticulture

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Abstract: The article presents a mathematical model and the results of dynamic calculation of hydraulic steering control of a four-wheel gantry high-clearance tractor, consisting of a distributor, pipelines and a steering mechanism – a hydraulic cylinder. The results of pressure and flow rate changes in the hydraulic cylinder are presented. The results of dynamic calculation can be used to study the process of turning movement of a tractor.

Keywords: gantry high-clearance tractor, turning radius, front axle, steering trapezoid, hydraulic steering gear, steering gear, angular velocity

1. INTRODUCTION

The area under orchards and vineyards is expanding every year. The areas allocated to dwarf orchards and vineyards, which have their own specific cultivation technology, are expanding more and more. Without proper mechanization of fruit cultivation it is impossible to obtain high productivity and labor efficiency in these areas.

In the Design and Technology Center for Agricultural Engineering, a design of steering mechanism for a gantry high-clearance tractor was proposed to ensure the wheels rotation of the front axle at a maximum angle (Figure 1). According to the requirements for these tractors, the machine turning should be within 5 ÷ 6 m.

A theoretical study of dynamic characteristics of the steering control (SC) is carried out using a mathematical model based on the element-node method [3]. According to this method, a complex system is divided into elements, the mathematical description of which is known, and the boundary conditions are set at the joint nodes of the elements.

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In order to expand the possible use of this SC in other machines, we have derived the analytical dependencies of kinematic parameters in a general form.

**Figure 1** - Steering gear: a - front view of a high clearance position; b – top view; c - front view of a low-clearance position

II. SYSTEM DESCRIPTION

Consider the kinematics of the steering trapezoid, which is the input mechanism for the SC (Figure 2).
Dynamic Calculation of a Steering Control of Gantry High-Clearance Tractor Used in Horticulture and Viticulture

Figure 2 - Kinematic diagram of the steering trapezoid of SC

As already has been noted, to build a mathematical model of SC, the element-node method was applied, i.e. each element of the SC (distributor, pipeline, steering gear – a hydraulic cylinder) was described mathematically and then connected according to the accepted circuit.

A study of the SC dynamics allows us to specify the selected design parameters of the SC elements, to determine the change in rotation radius in the headlands, the values of power consumption, the reliability of all systems connected with the process of turning.

For the study, a link performing a complex rotational-translational movement was taken as a reduction link.

We compose the equations of mechanism motion by reducing the masses and resistance forces to a single link. Let the rod of the hydraulic cylinder be the link of the reduction (Figure 2), i.e. the link \( l_2 = l_2(t) \). Then the dynamics of the hydraulic cylinder is described by the following system of differential equations [1, 7, 9]:

\[
\begin{align*}
\frac{d(m_{np}/2v_2^2)}{dt} &= P_d - k_v v_2 - P_{df} \sin v_2 - P_r \\
\frac{dp_1}{dt} &= \frac{E_f(\Omega - v_2 F)}{v_1}, \\
\frac{dp_2}{dt} &= \frac{E_f(V_2 F_1 - \Omega)}{v_1 + (l_{max} - l_2)/F_2}
\end{align*}
\]

where

\[
m_{np} = \sum_{i=2}^{6} m_i (v_1/v_2) + J_i (\omega_i/\omega_2)^2,
\]

\[
P_d = p_1 F_1 - p_2 F_2,
\]

\[
P_{df} = \sum_{i=2}^{6} P_i v \cos \phi_i/v_2 + M_i \omega_i/v_2
\]

\( m_{np} \) are the masses of moving parts brought to the cylinder rod; \( v_1, v_2, \omega_1 \) are the displacement, velocity and angular velocity of the hydraulic cylinder rod; \( k_v \) is the coefficient of viscous friction; \( P_{df} \) is the dry friction force; \( P_d \) is the driving force acting on the piston of the hydraulic cylinder; \( P_r \) is the resistance forces brought to the hydraulic cylinder rod; \( p_1, p_2 \) are the pressures in the head and drain cavities of the hydraulic cylinder; \( F_1, F_2 \) are the effective areas in the head and drain cavities; \( m_i \) is the mass of the \( i \)-th link; \( J_i \) is the moment of inertia of the \( i \)-th link relative to the axis passing through the center of mass; \( v_i \) is the velocity of the center of gravity of the \( i \)-th link; \( \omega_1 \) is the angular velocity of the \( i \)-th link; \( P_r, M_i \) are the magnitudes of active forces and moments acting on the links; \( \phi_i \) is the angle between the directions of the forces \( P_i \) and the velocity \( v_i \); \( l_i \) is the length of the \( i \)-th link.

The coordinates of the center of mass of the design in question can be determined by the formula

\[
\begin{align*}
x_c &= \sum G_i x_i / \sum G_i \\
y_c &= \sum G_i y_i / \sum G_i
\end{align*}
\]

Consider the masses and moments of links 3, 4, 5, 6 i.e. steering trapezoid, and the mass of the wheel, the front axle rack, the parameters of which are denoted below by the index 8.

Then the coordinates of the center of mass are determined by the formula [2]

\[
x_c = (m_3(x_{o3} + l_3 \cos \phi_3))/2 + m_4(x_{o4} + l_4 \cos \phi_4)/2 + m_5(x_{o5} + l_5 \cos \phi_5)/2 + m_6(x_{o6} + l_6 \cos \phi_6)/2 + m_7(x_{o7} + l_7 \cos \phi_7)/2 + m_8(x_{o8} + l_8 \cos \phi_8)/2 + m_9(x_{o9} + l_9 \cos \phi_9)/2 + m_{10}(y_{o10} + l_{10} \sin \phi_{10})/2 + m_{11}(y_{o11} + l_{11} \sin \phi_{11})/2 + m_{12}(y_{o12} + l_{12} \sin \phi_{12}) - m_{13}(y_{o13} + l_{13} \sin \phi_{13})/2 + m_{14}(y_{o14} + l_{14} \sin \phi_{14})/2 + m_{15}(y_{o15} + l_{15} \sin \phi_{15})/2 + m_{16}(y_{o16} + l_{16} \sin \phi_{16})/2
\]

Define the velocity and moment of inertia of the links as

\[
\begin{align*}
v_i &= \Omega_{o3}/2, \\
J_i &= m_i [(x_i - x_{o3})^2 + (y_i + y_{o3})^2] / 3 = 3.4, 5, 6, 8
\end{align*}
\]

where \( x_{o3}, y_{o3} \) are the coordinates of the center of gravity of the \( i \)-th link.

To determine the angular velocities, we take the derivatives from relations (2) and (5) [5, 8]:

\[
\begin{align*}
-l_3 \omega_3 \sin \phi_3 &= \frac{\dot{l}_2 \cos \phi_2 - \dot{\omega}_2 \sin \phi_2}{l_2} \\
-l_2 \omega_2 \sin \phi_2 &= \frac{l_2 \sin \phi_2 + \dot{\omega}_2 \cos \phi_2}{l_2} \\
-l_4 \omega_4 \sin \phi_4 &= \frac{-l_3 \omega_3 \sin \phi_3 - l_2 \omega_2 \sin \phi_2}{l_3} \\
l_4 \omega_4 \cos \phi_4 - l_2 \omega_2 \cos \phi_2 &= \frac{l_0 \omega_0 \cos \phi_0}{l_3} \\
l_4 \omega_4 \cos \phi_4 - l_2 \omega_2 \cos \phi_2 &= \frac{l_0 \omega_0 \cos \phi_0}{l_3}
\end{align*}
\]

Eliminating from (8) \( \omega_3 \) and \( \omega_2 \) we obtain:

\[
\omega_2 = \frac{l_2 \cos (\phi_3 - \phi_2)}{l_3 \sin (\phi_3 - \phi_2)}
\]

Eliminating from (9) \( \omega_6 \) and \( \omega_5 \) we obtain:

\[
\begin{align*}
\omega_{6,1} &= (-l_4 \omega_4 \cos \phi_1 - \phi_3 \pm \\
& \pm \sqrt{l_4^2 \omega_4^2 \cos^2 (\phi_2 - \phi_3) - l_2^2 (l_0^2 \omega_0^2 - \phi_2^2)/l_5^2} \\
\omega_{6,2} &= (-l_4 \omega_4 \cos \phi_1 - \phi_3 \pm \\
& \pm \sqrt{l_4^2 \omega_4^2 \cos^2 (\phi_2 - \phi_3) - l_2^2 (l_0^2 \omega_0^2 - \phi_2^2)/l_5^2}
\end{align*}
\]

where \( \omega_{6,1}, \omega_{6,2} = \omega_6 \).

To study the fluid flow in the pipeline, a model was chosen where the fluid is taken as compressible and concentrated in one or two volumes of small length (a system with concentrated parameters with account for the yielding of the hydraulic system elements). In this model, it is possible to take into account the compressibility of undissolved air bubbles [1, 7]

\[
\frac{dQ}{dt} = -\frac{f}{\rho} \frac{dp}{dx} - 27.5 \frac{hf^5}{\rho^2} Q - 0.443^{K_{e-x}}Q^2.
\]

\[
\frac{dp}{dt} = \left( \frac{E_p \delta_{p1} E_{pl}}{E_p \delta_{p1} E_{pl} + \delta_{p1} E_{pl}} \right) \frac{1}{f} \frac{dQ}{dx}.
\]
where \( p \) and \( Q \) are the pressure and fluid rate; \( t \) is the time; \( x \) are the coordinates along the axis of the circuit; \( \rho \) and \( E_y \) are the density and the volume modulus of fluid; \( d, \delta, E \) are the diameter, wall thickness and elastic modulus of the pipeline material, respectively; \( k_e \) is the approximation coefficient, the value of which depends on the relative roughness \( \varepsilon \) of hydraulic circuit; \( f \) and \( l \) are the area and the length of the pipeline, \( \mu \) is the fluid dynamic viscosity.

The flow rate through the distributor is determined by the dependence [3]:

\[
Q_p = \eta_p f_p(y) \sqrt{\frac{p_{pm} - p_{in}}{\rho \sin(p_{pm} - p_{in})}},
\]

where \( p_{pm} \) is the pressure created by pumps, \( f(y) \) is the area of the flow section, \( \mu \) is the flow coefficient.

The cross-sectional area of the distributor can be approximated by the following characteristic:

\[
\begin{align*}
0, & \quad t < \tau \\
\frac{\alpha d^2 t}{4}, & \quad t \leq t_k \\
\frac{\alpha d^2 t}{4}, & \quad t > t_k
\end{align*}
\]

where \( d \) is the nominal pipe size, \( \tau \) is the delay time, \( t_k \) is the time of full opening of the section.

The initial and boundary conditions for the problem in question are:

The initial conditions:

\( t=0, \quad p_{in}=10 \text{ MPa}, \quad p_{pm} = Q_{in} = p_{out} = Q_{out} \)

The boundary conditions:

- for a distributor:
  - at the inlet, the pump pressure \( p_{pm} \) is set;
  - at the outlet, the flow rate is \( Q_p \);
- for a pipeline of a length \( l \):
  - at \( x=0 \), \( Q_{in} = Q_{out} = 0 \), \( dp/dt = 0 \);
  - at \( x=l \), \( dp/dt = dV/dt = 0 \), \( p_{in} = p_{out} \);
- for a hydraulic cylinder:
  - at the inlet \( Q_{in} = Q_{out}(l) \), \( P_t \)
  - at the outlet \( p_1 \), \( p_2 \), \( v_2 \), \( l_2 \).

Thus, the system of equations (8-17), (20-22) together with the initial and boundary conditions is a mathematical model of the SC under consideration.

Consider the kinematics of the tractor turning and the forces acting on it at turning.

The instant center of machine turning can be found if the direction of the velocities of any two of its points is known [3, 4, 10].

Let the direction of velocities \( V_1 \) and \( V_2 \) of points 1 and 2 (Figure 3), which are the midpoints of the rear and front axles of the tractor two axes, be known; the direction of \( V_1 \) and \( V_2 \) is connected with several processes that occur under turning, in the absence of the pull or lateral sliding, the direction of velocities of each wheel would coincide with the planes of their rotation.

Lateral forces arising under tractor turning cause the wheel pull, which leads to a deviation of the directions of velocities \( V_1 \) and \( V_2 \) from the ones mentioned above.

Figure 3 – Scheme of tractor turning

The angles \( \delta_1 \) and \( \delta_2 \) to which the direction of the velocities \( V_1 \) and \( V_2 \) deviate due to the pull or lateral sliding, camber and suspension kinematics, will be called the pull angles of the front and rear axles, respectively. The ratio of the lateral force acting on the axle to the angle of pull is called the coefficient of resistance to the axle pull \( k_p \) for the front axle; \( k_p \) for the rear axle).

The instant center of machine turning is the point \( O_t \) of the intersection of perpendiculars to the directions of velocities \( V_1 \) and \( V_2 \). The radius of the turning is determined by expression [3, 4, 6]:

\[
R_T = \frac{R + 0.5}{\cos(\alpha - \delta)}
\]

where \( R = l/\big(tg(\alpha - \delta_1) + tg(\delta_2)\big) \); \( B \) is the base of machine.

The angular velocity of machine turning is determined by the expression:

\[
\omega_t = \frac{V_c}{R}
\]

\( V_c \) is directed along the longitudinal axis and is the machine velocity \( V_o \). Then:

\[
\omega_t = \frac{V_c}{R} = \frac{V_o(tg(\alpha - \delta_1) + tg(\delta_2))}{l}
\]

The location of the center of gravity is determined from the ratio:

\[
R_g = R_g b, b = l - a
\]

Hence: \( a = R_g l/(R_1 + R_2) \), \( b = l - a \)

The longitudinal \( P_x \) and transverse \( P_y \) components of the inertia forces in the coordinate system associated with the tractor have the form (Figure 4):

\[
\begin{align*}
P_x &= m_a (j - V_o \omega_o) \\
P_y &= m_a (V_a \omega_o + dV_o / dt)
\end{align*}
\]

where \( V_o, dV_o / dt \) are the velocity and acceleration of machine lateral displacement.

The positive direction \( P_x \) is opposite to the direction of machine movement and \( P_y \) is perpendicular to the direction of machine movement [7].

Lateral forces acting on the wheels are equal to the product of the coefficient of resistance to the pull \( k_p \) by the angle of pull \( \delta \):

\[
R_{\delta_t} = k_p \cdot \delta_1, R_{\delta_2} = k_p \cdot \delta_2
\]
The number of brake disks, the angles (\( t \) is written as \( \alpha \)), and the cosines are equal to unity (\( \delta_1 = 1 \)).

As an example, Figure 4 shows the dependence of lateral force on the load and the angles of pull.

**Figure 4 - Scheme of forces acting on the tractor**

The maximum value of lateral force is limited by the product of the load acting on the wheel by the coefficient of adhesion of the wheel to soil.

The resistance forces to wheel rolling on the front and rear axles along the road with a resistance coefficient \( f \) are determined as follows:

\[
R_{f1} = R_1 \cdot f, \quad R_{f2} = R_2 \cdot f;
\]

The braking force on each wheel under a disc brake is:

\[
R_{bi} = R_{B2} \cdot M_B \cdot n \cdot i_p / r_k
\]

where \( M_B \) is the braking torque developed by one disk, \( n \) is the number of brake disks, \( i_p \) is the gear ratio of the wheel gear, \( r_k \) is the radius of the wheel rolling.

The equation of lateral movement of the center of mass is written. Here, we assume that \( \delta_1 \) and \( \delta_2 \) are small and their cosines are equal to unity and the sines are equal to the angles (Figure 4)

\[
P_{sy} = R_{a1} \cos \alpha + R_{a2} \sin \alpha - (R_{B2} + R_{f2}) \sin \alpha
\]

or

\[
m_a \frac{dV_y}{dt} = \left( k_1 \delta_1 \cos \alpha + k_2 \delta_2 - (R_f + M_r / n_j) \sin \alpha \right) - V_y \omega_a
\]

The angles of pull \( \delta_1 \) and \( \delta_2 \) are expressed in terms of \( \alpha \), \( \omega_a \) and \( V_y \):

\[
\delta_1 = t g a - a \omega_a V_y / V_a, \quad \delta_2 = b \omega_a - V_y / V_a
\]

To obtain the equation of directional movement, we derive the equations of moments.

\[
J_c \frac{d\alpha}{dt} = -R_{a1} \alpha \cos \alpha + R_{a2} b + (R_{B2} + R_{f2}) \delta_2 \sin \alpha
\]

Or considering \( J_c = m a_b \)

\[
m_a \frac{d\alpha}{dt} = -k_1 \delta_1 \alpha \cos \alpha + k_2 \delta_2 b + (R_2 f + M_r / n_j) \delta_2 \sin \alpha
\]

Solving the equations, the dependence of the directional and lateral parameters of the wheeled vehicle movement on the angle of rotation of the steer wheels can be determined.

With known \( \omega_a \) and \( V_y \), we can find the coordinates of the center of mass of machine and its directional angle at each point in time [2, 7]:

\[
x = \int (V_a \cos \gamma - V_y \sin \gamma) dt + c_1
\]

\[
y = \int (V_a \sin \gamma + V_y \cos \gamma) dt + c_2
\]

\[
\gamma = \int \omega_a dt + c_3
\]

**III. SIMULATION RESULTS**

To solve the proposed mathematical models, an algorithmic program for numerical calculation was developed and implemented on a PC [7, 8, 9].

The flow rate of a metering pump was taken as an input impact for the numerical calculation of the system; it is expressed by the following dependence:

\[
Q_{mp} = \frac{\omega_s \cdot q_{mp}}{2 \cdot \pi}.
\]

where \( q_{mp} = 100 \text{sm}^3/\text{rev} = 1 \times 10^{-4} \text{m}^3/\text{rev}; \) \( \omega_s \) is the angular velocity of the steering wheel developed by a driver.

The trajectory of the tractor motion on the turns is mainly affected by such parameters as the angle of rotation of the guide wheels, the speed of motion, the total weight of machine, and the angles of pull of the guide and drive wheels.

180° turn of the tractor can be divided into two transient processes. In the first stage, the tractor moves from a straight section of the path \( R = \infty \) to a curved path of a constant least radius of curvature \( R = R_g (\gamma = 90°) \), and in the second stage, it moves from a curved path of a radius \( R = R_B \) to a straight section (\( \gamma = 180° \)).

Figures 6 - 11 show the dependences of changes in pull angles \( \delta_1 \) and \( \delta_2 \), the angular velocity \( \omega_a \), the lateral displacement velocity \( V_y \) of the tractor, the turning radius and the coordinates of the center of mass of the tractor (its motion trajectory).

In the calculations, the following parameter values were taken:

\( \rho = 850 \text{ kg/m}^2, \) \( E = 0.168 \times 10^{10} \text{ Pa, } d_{pl} = 0.012 \text{ m, } \delta_{pl} = 0.001 \text{ m, } E_{pl} = 1.5 \times 10^{12} \text{ Pa, } k_s = 0.023, \) \( L_s = 3.029 \text{ m, } L_T = 2.221 \text{ m, } v = 2.2 \text{ m/s, } l_{max} = 0.40 \text{ m, } d_f = 0.08 \text{ m, } d_r = 0.04 \text{ m, } k_s = 0.40 \text{ N/m, } P_{mp} = 200 \text{ N, } \mu_s = 0.5, \) \( t = 0.5 \text{ s, } d_{c1} = 0.01 \text{ m, } m_s = 350 \text{ N, } m_f = m_{pl} = 40 \text{ N, } m_r = 60 \text{ N, } m_t = 930 \text{ N, } m_w = m_{wheel} + m_{tractor} = 2450 \text{ N, } m_{total} = 12245 \text{ N, } F_1 = 1.96 \times 10^{-3} \text{ m}^2, \) \( F_2 = 2 \times 10^{-3} \text{ m}^2, \) \( F_{zz} = 1.34 \text{ m/s, } V_s = 1.59 \text{ m/s, } \omega_s = 6.28 \text{ rad/s, } m_s = 34986 \text{ N, } a_1 = 1.529 \text{ m/s, } a_2 = 0.001 \text{ m/s, } b_1 = 1.499 \text{ m, } b_2 = 1.056 \text{ m, } B_1 = 2.63 \text{ m, } c_1 = c_2 = x_0 = y_0 = 2 \text{ m.} \)
Figure 6 - Dependence of changes in angle $\delta_1$: 
(a) $L_а=3.029$ m; (b) $L_а=2.221$ m

Figure 7 - Dependence of changes in angle $\delta_2$: 
(a) $L_а=3.029$ m; (b) $L_а=2.221$ m

Figure 8 - Dependence of changes in angular velocity $\omega$: 
(a) $L_а=3.029$ m; (b) $L_а=2.221$ m

Figure 9 - Lateral displacement velocities $v_y$: 
(a) $L_а=3.029$ m; (b) $L_а=2.221$ m
IV. CONCLUSION

A mathematical model of the SC dynamics based on the element-node method is developed. The dependences of changes in the angles of pull, the angular velocity, the velocity of the tractor tank displacement, the radius of rotation and the coordinates of the center of mass of the tractor (the trajectory) are obtained.

It was established that for a high-clearance tractor with a base $L_u = 3.029$ m at velocity $V_p = 0.59$ m/s the radius of rotation was $R_p = 7.2$ m; at $V_p = 0.5$ m/s it was $R_p = 6.6$ m; at $V_p = 0.4$ m/s it was $R_p = 6.1$ m. For a tractor with a base $L_u = 2.22$ m, at velocity $V_p = 0.59$ m/s the radius of rotation was $R_p = 6.4$ m; at $V_p = 0.5$ m/s it was $R_p = 5.9$ m; at $V_p = 0.4$ m/s it was $R_p = 5.1$ m.

Thus, for a high-clearance tractor used in horticulture, the minimum safe turning radius is provided at a velocity of up to 0.4 m/s.

REFERENCES


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