International Journal of Innovative Technology and Exploring Engineering (IJITEE)
ISSN: 2278-3075 (Online), Volume-9, Issue-7, May 2020

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

Abdulaziz A. Shermukhamedov, Gulsara K. Annakulova, Bekzod J. Astanov, Sherzod A. Akhmedov, Yusufbek A. Shermukhamedov

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***

I. 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].

Revised Manuscript Received on May 30, 2020.
* Correspondence Author

Abdulaziz A. Shermukhamedov*, Tashkent Institute of design, construction and maintenance of automotive roads, Doctor of Technical Sciences, Professor of the Department of Road-building machines and equipment, Tashkent, Uzbekistan, E-mail: sheraziz@mail.ru

Gulsara K. Annakulova , Institute of Mechanics and Seismic Stability of Structures named after M.T. Urazbaev of the Academy of Sciences of the Republic of Uzbekistan, Candidate of Physical and Mathematical Sciences, Leading Researcher, Tashkent, Uzbekistan, E-mail: annaqulova_g@mail.ru

Bekzod J. Astanov , Institute of Mechanics and Seismic Stability of Structures named after M.T. Urazbaev of the Academy of Sciences of the Republic of Uzbekistan, doctor of philosophy (PhD) on technical sciences, Senior Researcher, Tashkent, Uzbekistan, E-mail: bekboy38@mail.ru

Sherzod A. Akhmedov, Design technology center of agricultural machinery LLC, doctor of philosophy (PhD) on technical sciences, Senior Researcher, Tashkent, Uzbekistan, E-mail: sheran@mail.ru

Yusufbek A. Shermukhamedov, Tashkent Institute of design, construction and maintenance of automotive roads, Magister, Tashkent, Uzbekistan, E-mail: Yusufbek.shermukhamedov@mail.ru

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.

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.

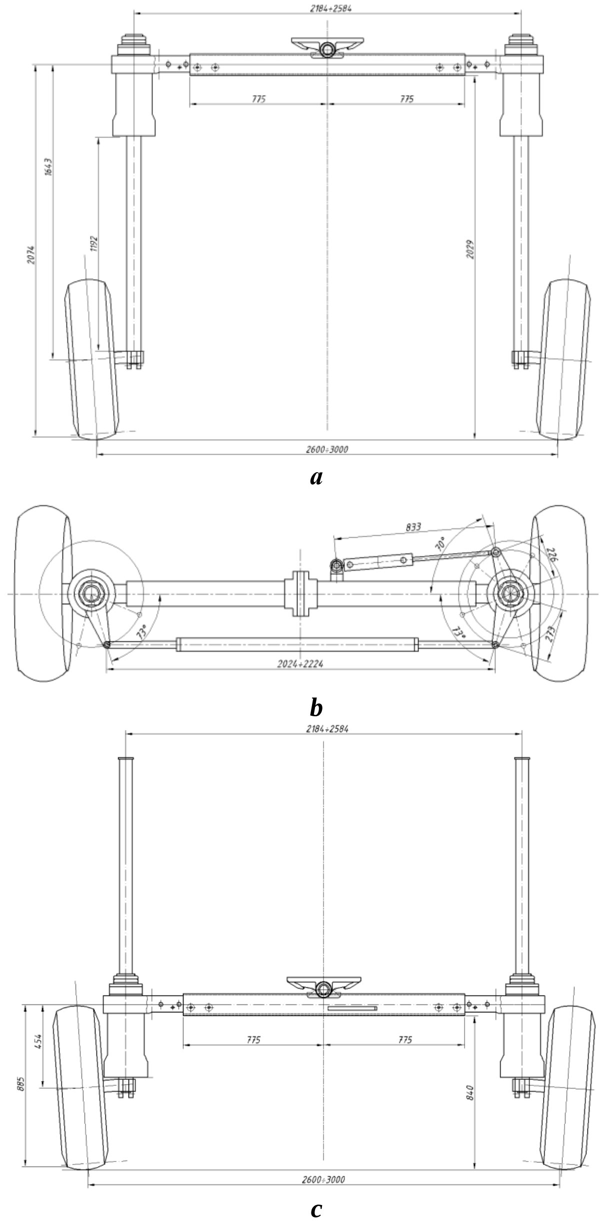


Figure 1 - Steering gear: a - front view of a high clearance position; b – top view; c - front view of a low-clearance position

II. SYSTEM DESCRIPTION

Consider the kinematics of the steering trapezoid, which is the input mechanism for the SC (Figure 2).

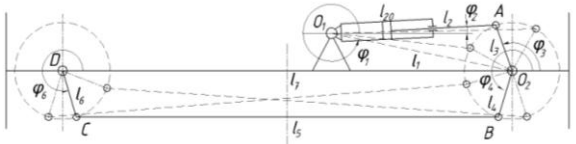


Figure 2 - Kinematic diagram of the steering trapezoid of SC

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 specific 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]:

$$\begin{cases} \dfrac{d\left(m_{br}/2v_2^{\,2}\right)}{dl_2} = P_d - k_v\, v_2 - P_{df}\sin v_2 - P_r \\ \dfrac{d\,\mathrm{p}_i}{dt} = \dfrac{E_f\left(Q_i - v_2F_i\right)}{v_{oi} + l_2F_i} \\ \dfrac{d\,\mathrm{p}_j}{dt} = \dfrac{E_f\left(v_2F_j - Q_j\right)}{v_{oj} + \left(l_{\max} - l_2\right)F_2} \end{cases} \tag{1}$$

where

$$m_{n\mathrm{p}} = \sum_{i=2}^{n} m_i\left(v_i/v_2\right) + J_i\left(\omega_i/\omega_2\right)^2,$$

$$P_d = p_1F_1 - p_2F_2,$$

$$P_c = \sum_{i=2}^{n} P_i v\cos\varphi_i / v_2 + M_i\omega_i / v_2$$

m_{br} – are the masses of moving parts brought to the cylinder rod; l_2, v_2, ω_2 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; ω_i is the angular velocity of the i-th link; P_i, M_i are the magnitudes of active forces and moments acting on the links; φ_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

$$\left.\begin{aligned} x_c &= \sum G_i x_i / \sum G_i \\ y_c &= \sum G_i y_i / \sum G_i \\ i &= 2 \ldots n \end{aligned}\right\} \tag{2}$$

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]

$$\begin{aligned} x_c = {} & (m_3\left(x_{02} + l_3\cos\varphi_3\right)/2 + m_4\left(x_{02} + l_4\cos\varphi_4\right)/2 + \\ & m_5\left(x_B + l_5\cos\varphi_5\right)/2 + m_6\left(x_D + l_6\cos\varphi_6\right) + m_7(x_D + \\ & + l_{12}\cos\varphi_6))/(m_3 + m_4 + m_5 + m_6 + m_8), \end{aligned} \tag{3}$$

$$\begin{aligned} y_c = {} & (m_3\left(y_{02} + l_3\sin\varphi_3\right)/2 + m_4\left(y_{02} + l_4\sin\varphi_4\right)/2 + \\ & + m_5\left(y_B + l_5\sin\varphi_5\right)/2 + m_6\left(y_D + l_6\sin\varphi_6\right) + m_7(y_D + \\ & + l_{12}\sin\varphi_6))/(m_3 + m_4 + m_5 + m_6 + m_8). \end{aligned} \tag{4}$$

Define the velocity and moment of inertia of the links as

$$v_i = l_i\omega_i/2, \tag{5}$$

$$J_i = m_i\left[\left(x_c - x_{ic}\right)^2 + \left(y_c + y_{ic}\right)^2\right],\ i = 3,4,5,6,8 \tag{6}$$

where x_{ic}, y_{ic} 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]:

$$\left.\begin{aligned} -l_3\omega_3\sin\varphi_3 &= \dot{l}_2\cos\varphi_2 - l_2\omega_2\sin\varphi_2 \\ l_3\omega_3\cos\varphi_3 &= \dot{l}_2\sin\varphi_2 + l_2\omega_2\cos\varphi_2 \end{aligned}\right\} \tag{7}$$

$$\left.\begin{aligned} -l_4\omega_4\sin\varphi_4 - l_5\omega_5\sin\varphi_5 &= -l_6\omega_6\sin\varphi_6 \\ l_4\omega_4\cos\varphi_4 + l_5\omega_5\cos\varphi_5 &= l_6\omega_6\cos\varphi_6 \end{aligned}\right\} \tag{8}$$

Eliminating from (8) ω_3 and ω_2 we obtain:

$$\begin{aligned} \omega_2 &= -\frac{\dot{l}_2\cos\left(\varphi_3 - \varphi_2\right)}{l_2\sin\left(\varphi_3 - \varphi_2\right)}, \\ \omega_3 &= -\frac{\dot{l}_2}{l_3\sin\left(\varphi_3 - \varphi_2\right)}. \end{aligned} \tag{9}$$

Eliminating from (9) ω_6 and ω_5 we obtain:

$$\begin{aligned} \omega_{5_{1,2}} &= (-l_4l_5\omega_4\cos\left(\varphi_4 - \varphi_5\right) \pm \\ & \pm\sqrt{l_4^2l_5^2\omega_4^2\cos^2\left(\varphi_4 - \varphi_5\right) - l_5^2(l_4^2\omega_4^2 - l_6^2)})/l_5^2 \\ \omega_{6_{1,2}} &= (-l_4l_5\omega_4\cos\left(\varphi_4 - \varphi_5\right) \pm \\ & \pm\sqrt{l_4^2l_5^2\omega_4^2\cos^2\left(\varphi_4 - \varphi_5\right) - l_5^2(l_4^2\omega_4^2 - l_6^2)})/l_6^2 \end{aligned}$$

where $\omega_4 = \omega_3$.

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]

$$\frac{dQ}{dt} = -\frac{f}{\rho}\frac{d\,\mathrm{p}}{dx} - 27{,}5\frac{\mu\lambda}{\rho f}Q - 0{,}443\frac{\kappa_\varepsilon\cdot\lambda}{f^{3/2}}Q^2, \tag{10}$$

Retrieval Number: G5798059720/2020©BEIESP
DOI: 10.35940/ijitee.G5798.059720
Journal Website: www.ijitee.org

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication

$$\frac{dp}{dt}=-\left(\frac{E_f\delta_{pl}E_{pl}}{E_{pl}\delta_{pl}+d_{pl}E_f}\right)\frac{1}{f}\frac{dQ}{dx}, \qquad (11)$$

where p and Q are the pressure and fluid rate; t is the time; x are the coordinates along the axis of the circuit; ρ and E_f are the density and the volume modulus of fluid; d_{pl}, δ_{pl}, E_{pl} are the diameter, wall thickness and elastic modulus of the pipeline material, respectively; k_ε is the approximation coefficient, the value of which depends on the relative roughness ε of hydraulic circuit; f and l are the area and the length of the pipeline, μ is the fluid dynamic viscosity.

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

$$Q_p=\eta_p f_p(y)\sqrt{2\left|p_{\mathrm{pm}}-p_{\mathrm{in}}\right|/\rho}\,\mathrm{sin}(p_{\mathrm{pm}}-p_{\mathrm{in}}), \qquad (12)$$

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

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

$$f(y)=\begin{cases}0, \text{ at } t<\tau\\ \pi d^2{}_y(t-\tau)/(4(t_k-t)), \text{ at } \tau\le t\le t_k,\\ \pi d^2{}_y/4, \text{ at } t>t_k\end{cases}$$

where d_y is the nominal pipe size, τ 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:

$t=0$, $\tau=0{,}2$, $t_k=2$, p_{pm} = 10 MPa, $p_{in} = Q_{in} = p_{ou} = =Q_{ou} = l_2 = v_2 = \omega_2 = p_1 = p_2 = 0$.

The boundary conditions:

- for a distributor:

at the inlet, the pump pressure p_{pm} is set;

at the outlet, the flow rate is Q_{fr};

- for a pipeline of a length l:

at $x=0$, $Q_{pl}=Q_{fr}$, $dp_{pl}/dt=0$

at $x=l$, $dQ_{pl}/dt=0$, $p_{pl}=p_i$;

- for a hydraulic cylinder:

at the inlet $Q_i=Q_{pl}(l)$, P_c

at the outlet p_i , p_j, 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 axles, 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.

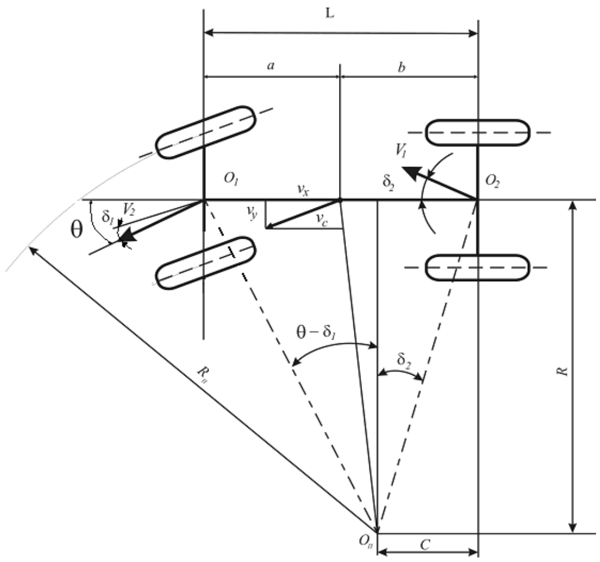


Figure 3 – Scheme of tractor turning

The angles δ_1 and δ_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_{y1} for the front axle; k_{y2} for the rear axle).

The instant center of machine turning is the point O_n 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]:

$$R_T=(R+0{,}5)/\cos(\alpha-\delta_1),$$

where $R=l/(tg(\alpha-\delta_1)+tg(\delta_2))$; B – is the rear wheels spacing; l is the base of machine.

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

$$\omega_a=V_C/R,$$

V_c is directed along the longitudinal axis and is the machine velocity V_a.

Then: $\omega_a=V_c/R=V_a(tg(a-\delta_1)+tg(\delta_2))/l.$

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

$$R_1a=R_2b, b=l-a$$

Hence: $a=R_2l/(R_1+R_2), b=l-a$

where R_1, R_2 – are the normal reactions of the road to the wheels of the front and rear axles.

The longitudinal P_{sx} and transverse P_{sy} components of the inertia forces in the coordinate system associated with the tractor have the form (Figure 4):

$$P_{sx}=ma(j-V_y\omega_a);$$

$$P_{sy}=m_a(V_a\omega_a+dV_y/dt);$$

where V_y, dV_y/dt are the velocity and acceleration of machine lateral displacement.

The positive direction P_{sx} is opposite to the direction of machine movement and P_{sy} 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 κ_y by the angle of pull δ:

$$R_{\delta 1} = k_{y1} \cdot \delta_1, R_{\delta 2} = k_{y2} \cdot \delta_2,$$

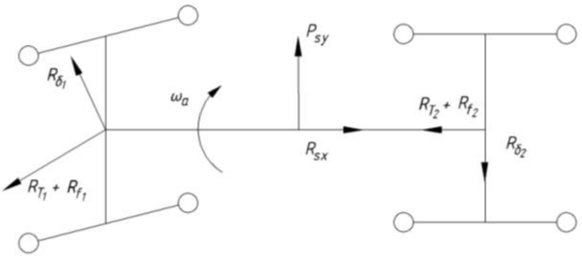


Figure 4 - Scheme of forces acting on the tractor

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.

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

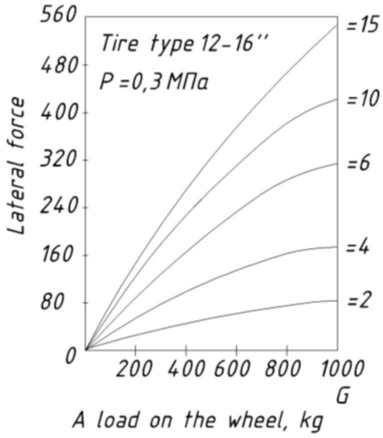


Figure 5 - Dependence of lateral force vs load

This dependence can be approximated by the following function:

$$k_y = 0.0016\delta^2 - 0{,}064\delta + 0.83$$

Longitudinal forces consist of resistance forces to the wheel rolling on the road and braking (traction) forces.

The resistance forces to wheel rolling of the front and rear axles along the road with a resistance coefficient *f* are determined as follows:

$$R_{f1} = R_1 \cdot f, \qquad R_{f2} = R_2 \cdot f;$$

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

$$R_{Bi} = R_{B2} = M_B \cdot n \cdot i_p / r_k$$

where $M_И$ 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 δ_1 are δ_2 are small and their cosines are equal to unity and the sines are equal to the angles (Figure 4)

$$P_{sy} = R_{\delta 1}\cos\alpha + R_{\delta 2} - (R_{B2} + R_{f2})\sin\alpha$$

or

$$m_a \frac{dV_y}{dt} = \begin{pmatrix} k_{y1}\delta_1\cos\alpha + k_{y2}\delta_2 - \\ -(R_1 f + M_T i_p / r_k)\sin\alpha \end{pmatrix} - V_a\omega_a$$

The angles of pull δ_1 and δ_2 are expressed in terms of α, ω_a and V_y.

$$\delta_1 = tg\alpha - \frac{a\omega_a + V_y}{V_a}, \quad \delta_2 = \frac{b\omega_a - V_y}{V_a}$$

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

$$J_z \frac{d\omega}{dt} = -R_{\delta 1} a\cos\alpha + R_{\delta 2} b + \\ +(R_{B2} + R_{f2}) b\sin\alpha$$

Or considering $J_z = m_a ab$

$$m_a ab \frac{d\omega}{dt} = -k_{y1}\delta_1 a\cos\alpha + k_{y2}\delta_2 b + \\ +(R_2 f + M_T i_p / r_k) b\sin\alpha$$

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 ω_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 qmp=100 sm3/rev = 1x10-4 m3/rev; ωs 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.

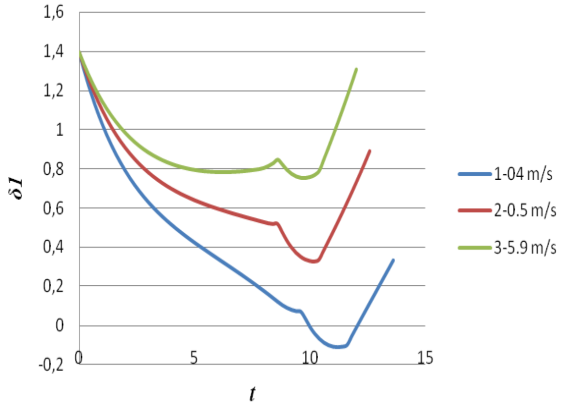
180° turn of the tractor can be divided into two transient processes. In the first stage, the tractor moves from a straight section of the path $R = \infty$ to a curved path of a constant least radius of curvature $R = R_B$ ($\gamma = 90°$), and in the second stage, it moves from a curved path of a radius $R = R_B$ to a straight section ($\gamma = 180°$). Figures 6 - 11 show the dependences of changes in pull angles δ_1 and δ_2, the angular velocity ω_a, the lateral displacement velocity V_v 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:

ρ=850 kg/m^2, E_f=0,168x10^{10} Pa, d_{pl}=0,012 m, δ_{pl}=0,001 m, E_{pl}=0,15x10^{12} Pa, k_ε=0,023, $L_н$=3,029 m, $L_в$=2,221m ν=2,2 1/s, l_{max}=0,40 m, d_s=0,08 m, d_{rod}=0,04 m, $k_в$=40 Nf/m, $P_{тр}$=200 N, μ=0,5, τ=0,5 s, d_y=0,01 m, m_3=350 N, m_4= m_6=40 N, m_5=60 N, m_7=930 N, m_8= m_{wheel}+m_{rack} =2450 N, $m_{пм}$ = 12245 N, F_1=1,96x10^{-3} m^2, F_2=7x10^{-4} m^2, $V_{нк}$ = 1,34 m/s, V_a = 1,59 m/s $\omega_{рк}$ = 6,28 rad/s, m_5=34986 N, $a_н$= 1,529 m, $a_в$= 1,1365 m, $b_н$ = 1,499 m, $b_в$ = 1,056 m B = 2,6÷3,0 m, c_1 = c_2 = x_0 = y_0 = 2 m.

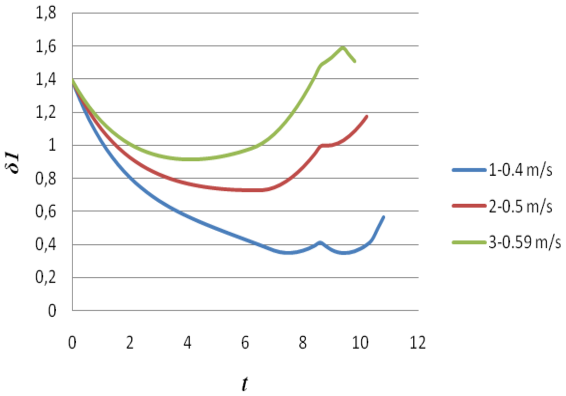
Retrieval Number: G5798059720/2020©BEIESP
DOI: 10.35940/ijitee.G5798.059720
Journal Website: www.ijitee.org

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication

International Journal of Innovative Technology and Exploring Engineering (IJITEE)
ISSN: 2278-3075 (Online), Volume-9, Issue-7, May 2020

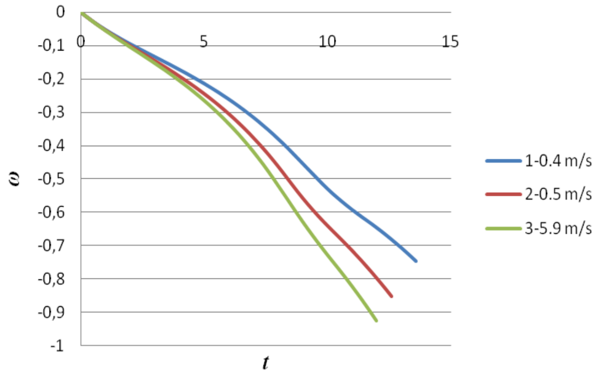


a

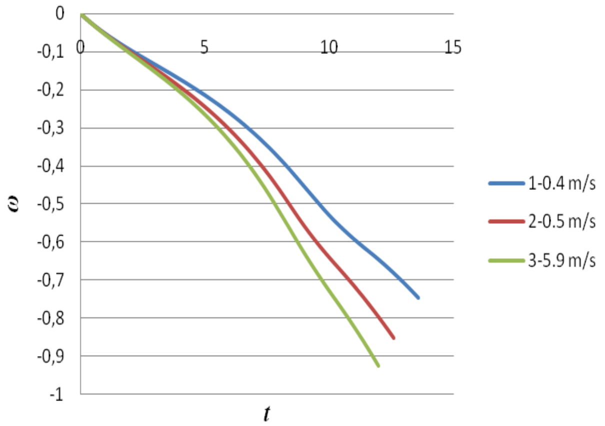


b

Figure 6 - Dependence of changes in angle δ_1: *a*) $L_н$=3.029 m; b) $L_в$=2.221 m

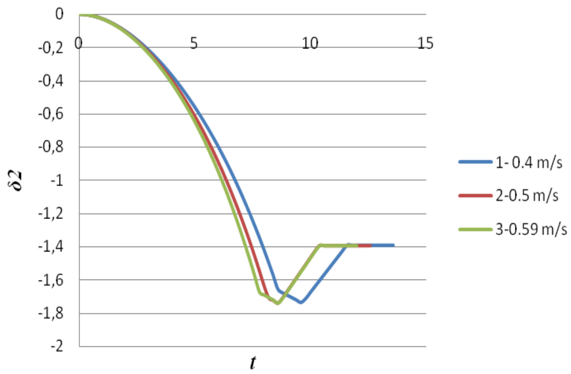


a

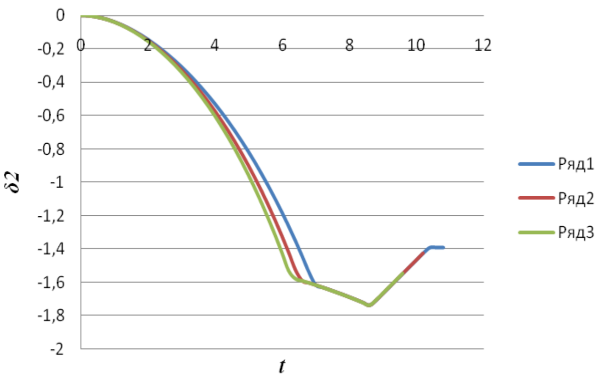


b

Figure 8 - Dependence of changes in angular velocity ω_a: *a*) $L_н$=3.029 m; *b*) $L_в$=2.221 m

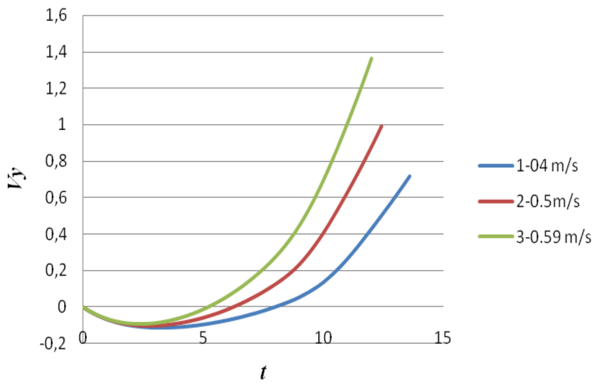


a

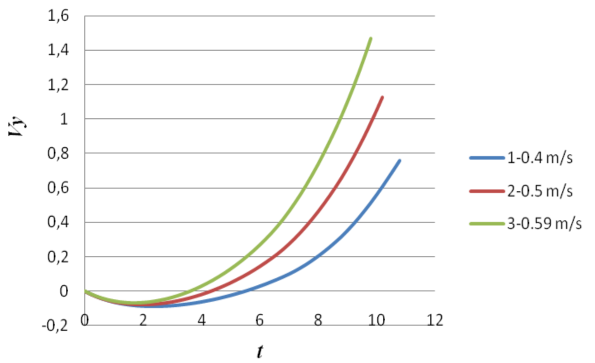


b

Figure 7 - Dependence of changes in angle δ_2: *a*) $L_н$=3.029 m; *b*) $L_в$=2.221 m

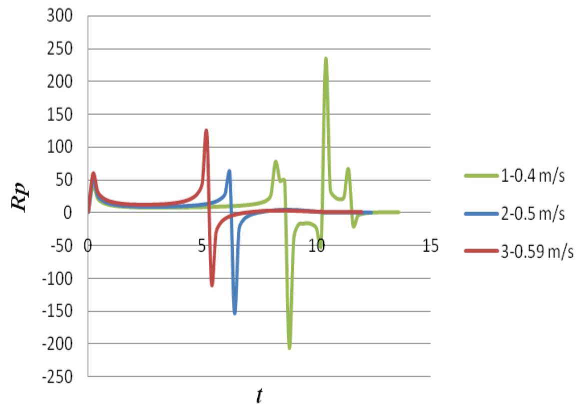


a

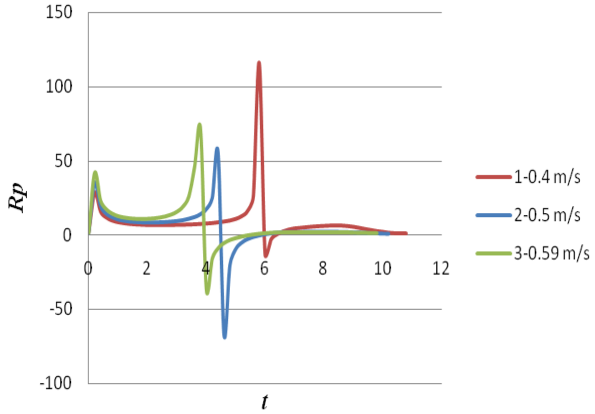


b

Figure 9 - Lateral displacement velocities v_y: *a*) $L_н$=3.029 m; *b*) $L_в$=2.221 m

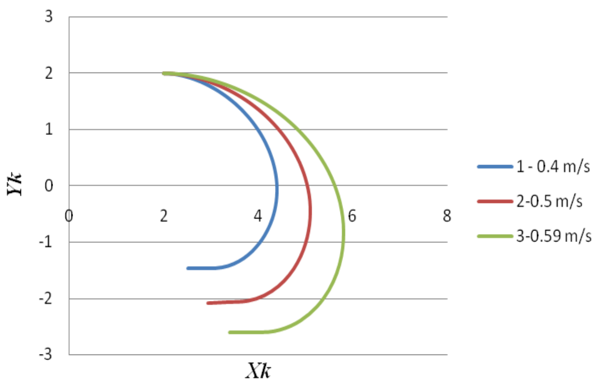


a

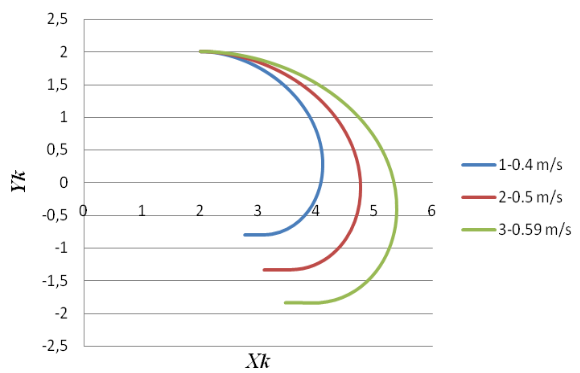


b

Figure 10 – Change in the radius of rotation of a tractor: *a*) L$_н$=3.029 m; b *)* L$_в$=2.221 m



a



b

Figure 11 - Dependence of changes in coordinates of the center of mass of tractor turning: *a*) L$_н$=3.029 m; *b)* L$_в$=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$_н$ = 3.029 m at velocity V_a=0.59 m/s the radius of rotation was R_n=7.2 m; at V_a.=0.5m/s it was R_n=6.6 m; at V_a.=0.4 m/s it was R_n=6.1 m. For a tractor with a base L$_в$=2.22m, at velocity V_a.=0.59m/s the radius of rotation was R_n=6.4m; at V_a.=0.5m/s it was R_n=5.9m; at V_a.=0.4m/s it was R_n=5.1m.

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

1. Metlyuk N.F., Avtushko V.P. The dynamics of pneumatic and hydraulic drives of cars. - M.: Mechanical Engineering, 1980 .-- 231 p.
2. Artobolevsky I.I. Theory of mechanisms. M. Nauka, 1967 - 719 p.
3. Guskov VV Tractors. Design and calculation. Part 3, Minsk Higher. school., 1981 – 380p.
4. Chudakov D.A. Fundamentals of the theory and calculation of the tractor and car. M., "Kolos", 1972
5. Zinoviev V.A. The course of the theory of mechanisms and machines. – M.: Nauka, 1975 .- 204 p.
6. Sharipov V.M. Designing and calculation of tractors. – Moscow: Machinery, – 2009. – P. 668-675.
7. Shermukhamedov A.A. Development of scientific foundations for modeling operation processes in hydraulic drives of freight mobile vehicles operating in extreme conditions. Diss ... Dr. Tech. Sci., - T., TADI. - 2000. - 265 p.
8. Shermukhamedov A.A., Annakulova G.K., Astanov B.J., Shermukhamedov Yu.A. Kinematics of the steering trapezoid of a four-wheeled energy-saturated universal-cultivating tractor TTZ-1033 // Vestnik TADI Scientific and Technical Journal, 2018. - No. 1. - P. 51-55.
9. Togaev A.A., Annakulova G.K., Astanov B.J., Shermukhamedov Yu.A. Dynamic calculation of hydraulic steering control of a four-wheel energy-saturated universal-cultivating tractor TTZ-1033 // Scientific and technical journal FerPI. - Fergana, No. 2, 2019. Volume 23. – P. 190-194.
10. Kambarov B.A. Method of calculating the kinematic parameters of steering gear ensuring a tractor minimum turning circle radius // European sciences review, Austria: Vienna, 2016 № 11–12 – p.131-134.

AUTHORS PROFILE

Abdulaziz A. Shermukhamedov, Tashkent Institute of design, construction and maintenance of automotive roads, Doctor of Technical Sciences, Professor of the Department of Road-building machines and equipment, Tashkent, Uzbekistan, E-mail: sheraziz@mail.ru

Gulsara K. Annakulova, Institute of Mechanics and Seismic Stability of Structures named after M.T. Urazbaev of the Academy of Sciences of the Republic of Uzbekistan, Candidate of Physical and Mathematical Sciences, Leading Researcher, Tashkent, Uzbekistan, E-mail: annaqulova_g@mail.ru

Bekzod J. Astanov *, Institute of Mechanics and Seismic Stability of Structures named after M.T. Urazbaev of the Academy of Sciences of the Republic of Uzbekistan, doctor of philosophy (PhD) on technical sciences, Senior Researcher, Tashkent, Uzbekistan, E-mail: bekboy38@mail.ru

Sherzod A. Akhmedov *, Design technology center of agricultural machinery LLC, doctor of philosophy (PhD) on technical sciences, Senior Researcher, Tashkent, Uzbekistan,
E-mail: sheran@mail.ru

Yusufbek A. Shermukhamedov, Tashkent Institute of design, construction and maintenance of automotive roads, Magister, Tashkent, Uzbekistan, E-mail: Yusufbek.shermukhamedov@mail.ru