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Analytical study of the turns of bridge machines

Abstract. The research is devoted to the topical problem of the efficiency of turning wide-span bridge machines in the track farming system. The research aims to study the curvilinear movement along the soil traces of a constant technological track of an arbitrary multi-supported bridge machine, considering its design and method of turning, parameters, modes of movement and loading. Experimental studies were conducted, involving the use of a modern strain track and specially designed equipment for electrical measurements of non-electrical quantities. The processing of research data was carried out on a personal computer. A methodology for compiling private models of turning off the bridge vehicle moving along the soil trace of the constant technological track was developed. As a result of the joint solution problem of the bridge machine turning, it is possible to determine all output parameters of curvilinear motion: trajectory, tractive forces, turning radius, slipping, and actual speeds. The force interaction of the bridge machine's undercarriage with the soil trace of a constant track is presented based on flat sliding with a variable anisotropic friction coefficient of adhesion φ_{yd}, depending on the properties of the track. As the radius of the wheel, the width of its tire and the air pressure in it, as well as the vertical load that acts on it, the coefficient of traction of the bridge machine φ_{yd} increases, which may cause higher slippage. The adequacy of the model of stationary turning of the overhead machine is confirmed by experimental estimation of the resistance coefficient of the power onboard turning. The convergence of the theoretical and experimental values of this

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coefficient is within the confidence interval ± σ in the whole investigated weight range of the bridge machine prototype. This allows us to assess the influence of design parameters and schemes on the turning characteristics as early as the design stage of a new bridge machine, thus optimizing the design process. The results of the research can be applied to the organization of the process of turning the existing models of bridge machines

Keywords: controlled traffic farming, gantry systems, curvilinear motion, coefficient of traction, coefficient of adhesion, front steered wheels

INTRODUCTION

The movement of bridge machines along the tracks of a permanent technological track creates somewhat different conditions and requirements for the operation of its pneumatic tire than for a traditional tractor moving along the agricultural background. As such, one of the requirements for the parameters of the technological track is their sufficient compaction, which improves the traction, coupling, and operational properties of bridge agricultural vehicles moving along them (Lou *et al*., 2021; Tamirat *et al*., 2022; Kørup *et al*., 2022). According to M.N. Thomsen *et al*. (2018), the restrictions on the permissible norms of the impact of undercarriage systems on the soil in the area of the tramline can be neglected. On the other hand, it has been established that the width of the bridge wheel should be as small as possible. This reduces the loss of field area under the technological zone.

Wide-span tractor (vehicles) for controlled traffic farming, like any vehicle, is a rather complex control object that can be adapted to manual or automatic control and is built according to the kinematic or power principle of turning. D.L. Antille *et al.* (2019) pointed out that kinematic rotation is implemented by turning the steered wheels (front, rear, or both front and rear) relative to the car frame. Wide-span tractors (vehicles) with steerable wheels are the most widely used.

S.Z.S. Al-khayyt (2018) presented studies aimed at optimizing the trajectory of curvilinear motion (PSO) by replacing the LSPB (linear segment with parabolic blend) trajectory when building an array of points on the trajectory. The PSO parameters were thus obtained to ensure that the LSPB trajectories accurately display the given waypoints. Interaction of the optimal trajectory of LSPB with PSO was demonstrated using mathematical modelling. The aforementioned method for the operational planning of the curvilinear turn of the bridge machine is quite simple and is used to solve such problems. However, the influence of the mass-geometric and structural-technological parameters of the wheeled vehicle on the process of its curvilinear movement is not considered.

M. Melnik *et al.* (2017) obtained parametric equations for the trajectory of the unsteady movement of a wheeled vehicle with front-steered wheels as a function of the angle of rotation of its body, which allows describing the processes of entering a left or right turn and exiting it. Using the models presented by the authors, it is possible to investigate the process of turning a wheeled vehicle as a function of time.

As such, many studies are devoted to the curvilinear movement of traditional wheeled and tracked vehicles, although, concerning bridge vehicles, this process has not been studied enough. The rotation of aggregates during fieldwork is also actively studied by scholars. A. Startsev *et al.* (2023) presented a mathematical model built to assess the rectilinear stability of the tractor unit in the uncontrolled turn mode. M. Song *et al*. (2013) attempted to optimize the point trajectory and simulate the creation of a given path for mowers as they turn. The authors proposed an algorithm based on experimental path-planning operation models for an autonomous mower. The results of production tests showed a certain convergence of the turning points.

Numerous global studies also aimed to study the movement of mobile units on a wheeled caterpillar mover. For example, M. Fashutdinov *et al*. (2020) considered the theoretical prerequisites for modelling the dynamics of the rotation of agricultural units with a wheel-caterpillar mover. The authors obtained mathematical models for determining the indicators of the turning of agricultural units. However, during modelling the process of curvilinear motion of a traditional tractor unit, the main assumption underlying the force interaction of the mover with the soil is the independence of the resulting force and friction moment. This limits the scope of models in this direction to large and medium turning radii and movement without loads.

The studies aimed to develop a model for turning a bridge machine moving along a compacted soil track of a constant tramline based on the theory of flat slip with a variable anisotropic coefficient of friction, which makes it possible to substantiate all the output parameters of its curvilinear movement.

MATERIALS AND METHODS

Experimental studies were carried out throughout 2020-2021 in a specially created laboratory for testing a bridge machine at Dmytro Motornyi Tavria State Agrotechnological University (Ukraine).

To conduct experimental studies, a four-wheeled overhead vehicle prototype was used (Fig. 1). Wheel track width was 3.5 m, base length was 2.3 m, and 9.5R32 tires were used.

Experimental studies were conducted in a specially equipped laboratory for testing with a test section length of 50 m.

Figure 1. Experimental bridge machine

Source: V. Bulgakov *et al*. (2021)

When conducting experimental studies, modern methods of strain measurement were used with a tense-resistive torque sensor. Transmission and fixation of strain gauge signals through transducers were transferred to a personal computer (PC). The results of strain measurement data were processed by statistical methods using a PC. The torque on the wheels of the axle machine was measured with a TW-2T-60K-S sensor, production – China, the characteristics of which are as follows:

Max Torque Capacity – 81 kN·m; Nonlinearity (% of Full-Scale Output) – 1%; Configuration – Heavy Duty Single or Dual Wheel.

The signal received from the torque sensor through an analogue digital converter was received and processed by a PC in the Excel software environment, discussed in detail by D. Zhuravel *et al*. (2020) and V. Borysov *et al*. (2020).

Modern methods of mathematical modelling of the behaviour of complex dynamic systems, agricultural machines and machine units were used. These methods involve modelling the movement of these objects by developing their equivalent circuits for both the entire machine-tractor unit and its element (wheel), as well as compiling differential equations of motion based on them. The resulting analytical dependencies were solved using well-known application programs on a PC. The results obtained are presented in the form of graphical dependencies.

RESULTS AND DISCUSSION

The operation of any wheeled vehicle occurs under many outside factors (forces and their moments), which change its position in space and deviate the movement from a given trajectory. The quality of processing one or another dynamic system of input variables depends on its characteristics. Relatively wide-span tractors (vehicles) such is its scheme, as well as design and other parameters. Therefore, the correct choice of the latter, from the standpoint of the necessary controllability and stability of its movement, provides it with the optimal transformation of the control and perturbing effects acting on it.

In the design of complex and expensive bridge machines (Fig. 2), the simulation process is becoming increasingly common, allowing to reduction, and sometimes eliminating, various types of in-situ testing (Pedersen *et al.*, 2016). In the operational cycle of any bridge machine a curvilinear motion (turning), the characteristics of which are often decisive in the design of new or evaluation of existing models of equipment, must be considered.

a) ASA-Lift WS 9600 WS b) ETC CTBE

Figure 2. Agricultural bridge machines

Currently, there are many steering models, most of which have been developed for fast-wheeled and tracked machines (Hac *et al.*, 2009; Shahgoli *et al.*, 2010; Nastasoiu & Ispas, 2014). The steering of such machines is adapted to manual steering and is built according to the kinematic or force principle of rotation. However, the application of these models to bridge wide-span machines is not always justified, due to the peculiarities of their technological cycle. Furthermore, some scientists recommend using exactly the power (onboard) rotation of the axle machine, which is implemented by rotating the wheels of the different sides of the machine at different speeds. Alternatively, combined kinematic-force diagrams are proposed for use, improving the controllability of overhead machines.

The specific features of the bridge machine sometimes pose additional problems in the study of curvilinear motion, not all of which have been solved until today. The idea of the scientific approach is to consider the bridge machine as a controlled object, the curvilinear motion of which is determined by the links imposed on it, provided by the design and control system, and interacting with the soil track of a constant technological track.

Traditional turning of wheeled machines is performed on a certain area, the size of which depends on the minimum radius of the turn, and includes several phases: entering the turn, the turn itself and exiting from the turn (Nadykto *et al.*, 2015). Due to the presence of elasticities in the "wheel–supporting surface" system, the trajectory of the mobile machine when turning does not depend unambiguously on the turn of the steering wheel. All this makes it difficult to automate the process of turning (Nastasoiu & Ispas, 2014). The most widespread methods for turning mobile machines are those that involve rotating the steered wheels (front, rear or both front and rear) relative to the body or by changing the position of one part of the mobile machine relative to the other in the horizontal plane (articulated frame). In this case, if the machine has all the steering wheels, they can turn simultaneously on the front and rear axles in different directions (full-return mode or "path in the track") or to one side (crab mode). Considering the general structure of its chassis and steering drive, a purely on-board (power) turn in the horizontal plane, which is carried out by different rotation speeds of the wheels of the right and left side, is considered a promising way to turn the overhead machine. In this case, a four-wheeled axle machine with a sideways turn is turned around the absolute centre of turn of the intersection of the extension of the front and rear axles. The lateral displacement of the machine relative to the trajectory line is possible only as a result of uncontrolled sideways drift.

There are two main theoretical methods for wheeled machine rotation in terms of describing the force interaction between the propulsor and the supporting surface.

First – for machines with steerable wheels and articulated frames, the lateral departure method is widely used (Pascuzzi, 2015). The description of force interaction is based on the Rocard hypothesis of linear dependence of lateral force on the angle of wheel departure, explained by the elastic properties of the tire and applicable in the absence of sliding (Wang *et al.*, 2016). Attempts to account for wheel slip of a machine within the theory of lateral guidance run counter to the underlying assumptions, and sometimes violate the laws of mechanics. Models of this type have proven themselves well in describing the rotation of fast cars on a solid base with large turning radii, where there is virtually no slip of the wheels and, the transverse force can be easily explained by the action of the normal acceleration. However, the movement of a bridge vehicle with wheels that do not have large hooks, due to their lack of use when driving on the constant hard track, will inevitably slip when turning. Therefore, models based on lateral guidance cannot be applied to describe the curvilinear motion of a bridge machine.

Another type of mobile machine with non-swivelling wheels and power sideways rotation, where the interaction of the propulsor with the ground is characterized by increased slip (Wang *et al.*, 2008). The elastic properties of the tire are neglected in these models. Following those, the basic forces in contact are reduced to a single equilibrium force applied at the centre of pressure, which corresponds to progressive sliding, incompatible with curvilinear motion. Adding a friction torque to the result traction limit will exceed the traction limit of the machine. When a machine turns, with the friction torque present, the value of the resultant force must be less than its traction limit. The larger the torque, the smaller the final force, which makes it unpredictable and does not allow the ground reaction to be explicitly noted.

Different methodological approaches in describing the interaction of wheeled and tracked movers with the soil complicate their application to a bridge vehicle, especially with a combined mover. In the theory of turning of wheeled and tracked machines, models of passive turning, which takes place in the technological cycle of many bridge machines, are insufficiently studied.

At the beginning of the study, the assumption that the turning of the bridge machine, regardless of the type of its mover, is accompanied by sliding within the soil trace of a constant track was considered. The elementary tangential friction (adhesion) forces caused by the contact of the propulsor with the bearing surface of the soil track of a constant technological track are determined by the direction of relative motion and the anisotropy of the interaction between the propulsor and the soil. In this case, in the process of turning the bridge machine, its propulsor performs a flat motion (Fig. 3).

At each point of contact of the axle-motor with the constant track (Fig. 3), a basic friction force $dT = \varphi \cdot q \cdot dF$ (where φ – adhesion coefficient, q – the normal pressure at the elementary site dF of the bearing surface) is present, directed in the opposite direction to the linear velocity V, which in turn is perpendicular to the radius r, connecting the point with the instantaneous centre of the sliding (adhesion) velocities.

Combining all basic forces (Fig. 3) to the instantaneous centre of velocities (point C), and adding them over the contact area, expressions for the projections of the resultant force (in the longitudinal T_y and transverse T_x directions) and the moment M are obtained:

$$\left.\begin{aligned} T_x &= -\int_\eta \int_\xi q\varphi_x \frac{y-\eta}{\sqrt{(x-\xi)^2+(y-\eta)^2}} d\xi\, d\eta, \\ T_y &= \int_\eta \int_\xi q\varphi_y \frac{x-\xi}{\sqrt{(x-\xi)^2+(y-\eta)^2}} d\xi\, d\eta, \\ M &= \int_\eta \int_\xi q\left(\frac{\varphi_x(y-\eta)^2+\varphi_y(x-\xi)^2}{\sqrt{(x-\xi)^2+(y-\eta)^2}}\right) d\xi\, d\eta, \end{aligned}\right\} \quad (1)$$

where q – the normal pressure at the contact point with coordinates ξ, η; φ_x, φ_y – transverse and longitudinal coefficients of adhesion at the point with coordinates ξ, η.

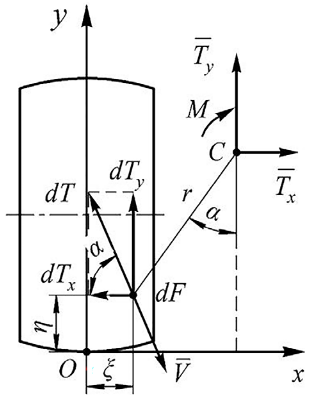

Figure 3. Formation of forces in the contact of the wheel of a bridge machine with the constant track
Note: dT – basic friction force; **φ** – friction coefficient; q – normal pressure; dF – elementary area of the support surface; V – linear velocity; r – radius connecting the point with the instantaneous centre of sliding (adhesion) velocities; C – instantaneous centre of velocities; T_x, T_y – projections of the resulting force in the transverse and longitudinal directions; M – moment; **α** – angle of inclination of the elementary friction force in the longitudinal direction
Source: developed by the authors

Dependence (1) considers the relationship between the force and the friction (adhesion) moment, as well as the anisotropy of the interaction through different adhesion coefficients φ_x, φ_y. The force factors in it are formed in the contact of each propulsor support with the soil. The integration limits can be used for any shape and size of the contact surface. However, in describing the rotation of a bridge vehicle, dependence (1) does not consider the deformation of the soil in the traces of the constant technological track and the elastic properties of pneumatic tires.

Elastic properties of the soil trace of a constant track will be considered by variable coefficients φ_x, φ_y. To determine them, the maximum tangential traction force, which is developed by the wheel of a bridge machine, is assumed to be:

$$P_{k\max} = \varphi_{yd} \cdot N_{ek}, \tag{2}$$

where φ_{yd} – longitudinal coefficient of traction at full slip, N_{ek} – vertical operating load, which acts on the wheel of the bridge machine. From tractor theory, it is known that the maximum tangential traction force that a tractor wheel develops can be determined from the expression (Nadykto *et al.*, 2020):

$$P_{k\max} = \delta_{\max} \cdot S_k \cdot k_0 \cdot L, \tag{3}$$

where $\delta_{\max}$ – slip ratio (maximum) of the axle machine wheels, S_k – sum of the vertical projections of the bearing surfaces of the wheel sprocket submerged in the soil, k_0 – sum of the vertical projections of the bearing surfaces of the wheel sprocket submerged in the soil, L – is the length of the arc of traction of the wheel hitches with the bearing surface (the trace of the permanent tramline), the value of which find from the expression (Nadykto *et al.*, 2020):

$$L = r_k \cdot \left(\arctan \frac{f_k \cdot (1-f_k^2)^{\frac{1}{2}}}{0.5 - f_k^2} + 2 \cdot f_k^2 \right), \tag{4}$$

where r_k – wheel radius, f_k – wheel rolling resistance coefficient.

The rolling radius r_k of the wheel of a bridge machine can be determined by its static radius r_0 and normal tire deflection h_z:

$$r_k = r_0 + h_z. \tag{5}$$

The normal tire deflection of a mobile machine or tractor wheel can be determined from the expression (Mitkov *et al.*, 2021):

$$h_z = \frac{N_{ek}}{\pi \cdot \rho_w \cdot \sqrt{2 \cdot r_0 \cdot b_0}}, \tag{6}$$

where ρ_w – tire pressure, r_0, b_0 – static radius and wheel diameter of the bridge machine.

The sum of the vertical projections S_k of the bearing surfaces of the wheel axles of the bridge machine, buried in the soil, determined from the expression (Mitkov *et al.*, 2021):

$$S_k = \pi \cdot h_z \cdot [(2 \cdot r_0 - h_z) \cdot (b_0 - h_z)]^{\frac{1}{2}}. \tag{7}$$

By substituting the values of S_k, L, r_k (4-7) into (3) and equating the expressions (2) have:

$$\delta_{\max} \cdot \pi \cdot h_z \cdot [(2 \cdot r_0 - h_z) \cdot (b_0 - h_z)]^{\frac{1}{2}} \cdot (r_0 + h_z) \left(\arctan \frac{f_k \cdot (1-f_k^2)^{\frac{1}{2}}}{0.5 - f_k^2} + 2 \cdot f_k^2 \right) = \varphi_{yd} \cdot N_{ek}. \tag{8}$$

From the analysis of the obtained equation (8) it follows that with the increase of the coefficient of adhesion φ_{yd} of the bridge machine under the contact of its wheels with the bearing surface of the constant track, the maximum slip coefficient $\delta_{\max}$ increases. With the increase of the wheel radius r_0, the width of the tires b_0 and air pressure ρ_w, as well as the vertical load N_{ek}, exerted, the wheel grip φ_{yd} with the bearing surface of the permanent technological track increases as well. Hence the conclusion is that the greater the traction of the wheel of the bridge machine with the bearing surface of the constant technological track, the more slipping can be achieved. The nature of this relationship is determined by the parameters of the bridge machine, the mode of its movement and the properties of the bearing surface of the traces of a constant track.

Results of studies of the maximum slip coefficient estimation δ_{max} of the wheel of a bridge machine from the value of its friction coefficient φ_{yd} are shown in Figure 4.

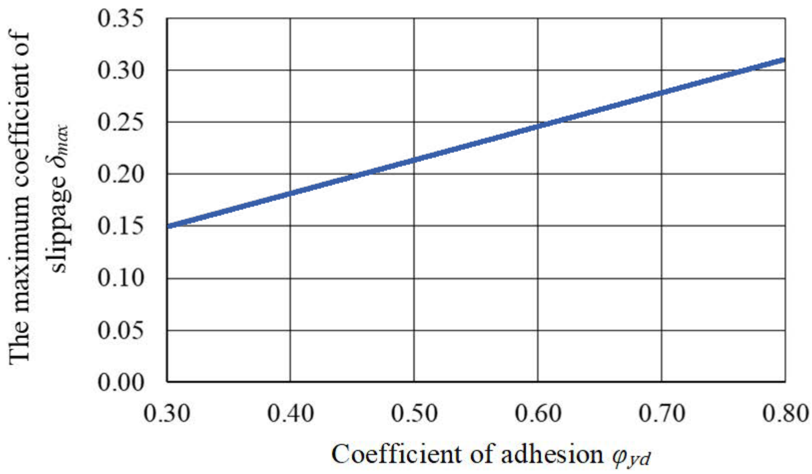

Figure 4. Dependence of the maximum slipping ratio of the bridge machine δ_{max} from the coupling coefficient φ_{yd} with the bearing surface of the soil trace of the permanent technological track

Note: δ_{max} – slip coefficient; φ_{yd} – adhesion coefficient

Source: developed by the authors

The result obtained in Figure 4 can be explained as follows. The phenomenon of slipping of the bridge machine wheels is caused by the displacement of the support surface of the trace of the constant track by the tires until the necessary amount of tangential stress is formed in it. The higher the strength of the bearing surface on which the wheel of the bridge machine moves, the less the slippage affects the connection between the soil particles. As a result, the depth of the wheels sinking into the soil is reduced and, accordingly, the energy consumption for the formation of a groove by the star-shaped hooks is reduced. Therefore, the degree of wheel slip is determined by the amount of horizontal deformation (shearing) of the soil, which is carried out by the star gears. The specified horizontal deformation of the soil depends on the specific pressure on the bearing surface of the permanent trace of the technological track and on its ability to resist deformation.

The force factors T_x, T_y, M in the contact of the wheel of the bridge machine with the bearing surface of the constant trace of the technological track depend on the unknown coordinates of the instantaneous velocity centre, which reduces the force problem to the kinematics of its motion. Next, it was investigated the kinematics of the curvilinear motion of the constant track system – mover–bridge machine, and describe the connections arising in the process of its rotation. It was introduced a stationary coordinate system (xOy), the machine's motion system ($x_1O_1y_1$) and local frames of reference (x_{ij} y_{ij}), associated with each of the thrusters of the bridge machine (Fig. 5).

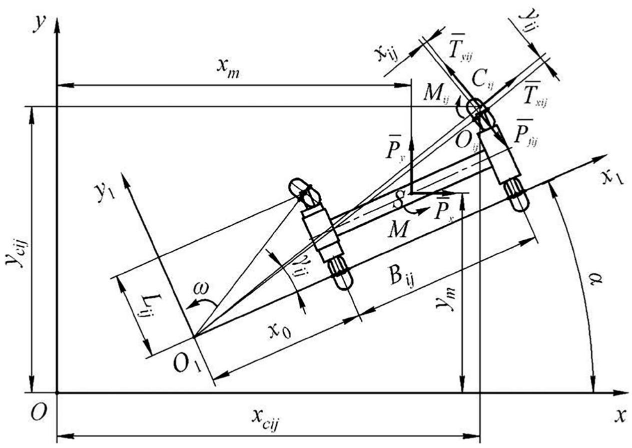

Figure 5. Diagram of unsteady rotation of an arbitrary multi-support bridge machine

Note: xOy – system of fixed coordinates; $x_1O_1y_1$ – system of moving coordinates coinciding with the centre of rotation of the machine; x_m, y_m, α – current coordinates of the centre of mass and the angle of rotation of the machine body in the fixed coordinate system; P_{fij} – resistance to movement ij of the supporting wheel; m, J – mass and moment of inertia of the machine relative to the vertical axis passing through its centre of mass; P_x, P_y, M – external forces and external moment, reduced to the centre of mass of the machine; T_{xij}, T_{yij}, M_{ij} – force factors in contact ij of the support wheel with the trace of the constant technological track; γ_{ij} – angle of rotation ij of the support relative to the vertical axis in the machine system

Source: developed by the authors

To describe the unsteady motion of an arbitrary multi-support bridge machine with n axes ($i = 1, ..., n$) and m supports on each axis ($j = 1, ..., m$) a generalized model of its controlled rotation should be formulated:

$$\left.\begin{aligned} m\cdot\ddot{x}_m &= \textstyle\sum_{i=1}^{n}\sum_{j=1}^{m}\left[T_{xij}\cdot\cos(\gamma_{ij}+\alpha) - T_{yij}\cdot\sin(\gamma_{ij}+\alpha) + f_{ij}\cdot G_{ij}\cdot\sin(\gamma_{ij}+\alpha)\right] + P_x,\\ m\cdot\ddot{y}_m &= \textstyle\sum_{i=1}^{n}\sum_{j=1}^{m}\left[T_{xij}\cdot\sin(\gamma_{ij}+\alpha) + T_{yij}\cdot\cos(\gamma_{ij}+\alpha) - f_{ij}\cdot G_{ij}\cdot\cos(\gamma_{ij}+\alpha)\right] + P_y,\\ J\cdot\ddot{a} &= \textstyle\sum_{i=1}^{n}\sum_{j=1}^{m}\left[T_{yij}\cdot\sin\gamma_{ij} - T_{xij}\cdot\cos\gamma_{ij}\cdot(L_{ij} - y_m + x_{ij}\cdot\sin\gamma_{ij} + y_{ij}\cdot\cos\gamma_{ij}) - \right.\\ &\quad -f_{ij}\cdot G_{ij}\cdot\cos\gamma_{ij}\cdot(B_{ij} - x_m) - f_{ij}\cdot G_{ij}\cdot\sin\gamma_{ij}\cdot(L_{ij} - x_m) - M_{ij} +\\ &\quad \left. +(T_{yij}\cdot\cos\gamma_{ij} + T_{xij}\cdot\sin\gamma_{ij})\cdot(B_{ij} - x_m + x_{ij}\cdot\cos\gamma_{ij} - y_{ij}\cdot\sin\gamma_{ij})\right] + M, \end{aligned}\right\} \tag{9}$$

where x_m, y_m, α – the current coordinates of the centre of mass and the angle of rotation of the machine body in a fixed Cartesian system, $f_{ij} \cdot G_{ij} = P_{fij}$ – resistance of support wheel movement ij, m, J – mass and moment of inertia of the machine relative to the vertical axis passing through its centre of mass, P_x, P_y, M – external forces and external momentum applied to the centre of mass of the machine, T_{xij}, T_{yij}, M_{ij} – force factors in the contact of the support wheel ij with a constant technological track. The initial conditions of motion of the bridge machine (coordinates of the instantaneous velocity centre) are taken from the solution of the moving-away problem. Next, it was written down the coupling reactions of the bridge machine for the considered diagram (Fig. 5). In the case of the power (onboard) rotation of the bridge machine, when the supports of its thrusters are unguided, it is sufficient to consider its stationary rotation, according to which the beginning of the moving system $x_1O_1y_1$ coincides with the turning centre of the turning machine. In this case, the kinematics of the unguided randomly positioned support wheel of the bridge machine is shown in Figure 6. Based on Figure 6, the actual speed V_{Dij} of the contact patch points (point A_{ij}) of each arbitrarily positioned support wheel ij of a bridge machine is the sum of the theoretical speed V_{Tij} and sliding speeds V_{CKij}.

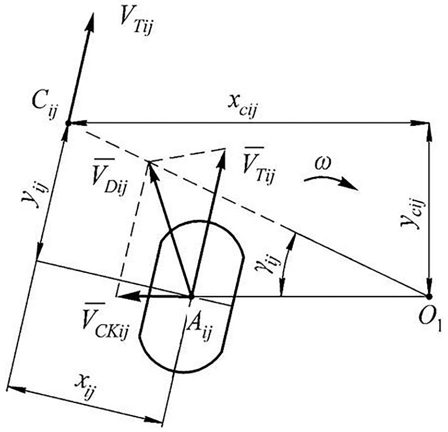

Figure 6. Kinematics of an unguided randomly positioned support wheel of a bridge machine

Note: C_{ij} – point of the wheel body located above it; V_{Tij} – theoretical speed of movement directed along the rolling plane ij of the support wheel; A_{ij} – point of the contact spot of any ij arbitrarily located support wheel of the bridge machine; V_{Dij} – actual speed of the contact spot point; V_{Tij} – theoretical speed; V_{CKij} – sliding speed; x_{cij}, y_{cij} – coordinates of the instantaneous centre of speeds of the propeller support in the machine system; V_{Tij} – theoretical speed of the axis ij of the support; O_1C_{ij} – distance from the centre of rotation of the machine to the instantaneous centre of speeds ij of the support; γ_{ij} – angle of rotation ij of the support relative to vertical axis in the machine system

Source: developed by the authors

The only point in Figure 6 without slip is the instantaneous velocity centre, therefore the wheel body point (C_{ij}), located above it has only theoretical velocity V_{Tij}, directed along the rolling plane of the support wheel ij. Using the orthogonality theorem for the unguided thruster support, according to which, when the machine rotates, the instantaneous centre of velocities of the support area of an arbitrarily located thruster support lies on the perpendicular dropped from the centre of rotation of the machine to the plane of its rolling, write the bond equations for each support wheel in the form:

$$x_{cij} \cdot \sin_{\gamma ij} - ycij \cdot \cos\gamma_{ij} = 0, \tag{10}$$

and

$$V_{Tij} = \omega \cdot (O_1C_{ij}), \tag{11}$$

where x_{cij}, y_{cij} – coordinates of the instantaneous velocity centre of the thruster support in the machine system, V_{Tij} – theoretical speed of the ij axis of the support, O_1C_{ij} – distance from the centre of rotation of the machine to the instantaneous centre of velocities ij of the support, γ_{ij} – rotation angle ij of the support relative to the vertical axis in the machine system.

The geometric relation equation (10) reflects the design scheme and parameters of the bridge machine (base, track, number of supports and their mutual positioning) since the transformation of coordinates of the instantaneous centre of speeds (x_{cij}, y_{cij}) for any design of the support and running system (with steerable or nonrotating support wheels) is performed by consecutive shifting and turning.

The kinematic relationship equation (11) is determined by the turn control scheme and reflects the mode of motion of each support (driven, leading, braking). In some cases, it is more convenient to write equation (11) through force factors, since the latter are functions of instantaneous velocity centre coordinates:

- driven wheel (flange) of a bridge machine:

$$T_{yij}=0, \tag{12}$$

- brake wheel (side) of an overhead machine:

$$x_{cij}=0, \tag{13}$$

- individually driven motor-wheel:

$$\omega = \frac{V_{Tij}}{\sqrt{x_{cij}^2+y_{cij}^2}}, \tag{14}$$

- individually driven motor-wheel:

$$k = \frac{V_{mj+1}}{V_{mj}} = \frac{x_0+B+x_{j+1}}{x_0+x_j}. \tag{15}$$

A different combination of geometric (10) and kinematic equations (12-15) allows for a description of restrictions imposed by the design scheme of the bridge vehicle (number of propulsor supports, their mutual arrangement, steering scheme, base, and track) and control system on its curvilinear motion parameters. As a result of solving the system of equations (9) considering the unknowns x_m, y_m, α and the coordinates of the instantaneous velocity centre (x_{ij}, y_{ij}), all force and kinematic characteristics of curvilinear motion of the bridge machine were acquired, namely angular velocity, trajectory of motion, longitudinal and transverse components of the velocity of the centre of mass, actual turning radius, normal acceleration, traction forces and slipping on the leading supports, distribution of normal reactions on the supports, power losses and their change in time, depending on the control parameters.

Considering that when turning, the bridge machine moves within the boundaries of the trace of a constant track, its curvilinear movement must occur with a constant radius. Therefore, the generalized model of controlled curvilinear motion requires a transformation to build partial models of the system solution (9). For this, it is enough to consider the stationary and statistical stationary modes of the turning of the bridge machine. A stationary turn is characterized by a constant radius and acceleration of the bridge machine and is realized with constant control parameters:

$$\ddot{\gamma}_{ij}=0,\ \ddot{\alpha}=0. \tag{16}$$

The model of the stationary rotation is made in the natural coordinates of the moving trihedron ($\alpha=0$). The unknowns are the coordinates of the centre of rotation in the machine system of its angle speed y_0, its angle speed ω and the coordinates of the instantaneous velocity centre x_{ij}, y_{ij}. The stationary rotation model is a system of $2nm$ coupling equations (10-15) and three equations of motion:

$$\left.\begin{aligned} -m\cdot\omega^2\cdot y_0 &= \textstyle\sum_{i=1}^n\sum_{j=1}^m\left(T_{xij}\cdot\cos\gamma_{ij} - T_{yij}\cdot\sin\gamma_{ij} + f_{ij}\cdot G_{ij}\cdot\sin\gamma_{ij}\right) + P_x,\\ -m\cdot\omega^2\cdot x_0 &= \textstyle\sum_{i=1}^n\sum_{j=1}^m\left(T_{xij}\cdot\sin\gamma_{ij} + T_{yij}\cdot\cos\gamma_{ij} + f_{ij}\cdot G_{ij}\cdot\cos\gamma_{ij}\right) + P_y,\\ 0 &= \textstyle\sum_{i=1}^n\sum_{j=1}^m\left[-M_{ij} + T_{yij}\cdot\sqrt{x_{cij}^2+y_{cij}^2} - f_{ij}\cdot G_{ij}\cdot\left(\sqrt{x_{cij}^2+y_{cij}^2} - x_{ij}\right)\right] + M. \end{aligned}\right\} \tag{17}$$

Static stationary rotation is characterized by low operating speeds ($\omega=0$) of the bridge machine, with the centrifugal forces being so small that they can be neglected. This reduces by one the number of unknowns and converts equations of motion (17) to equilibrium equations (the left parts are zero). The number of geometric coupling equations (10) is equal to the number of bridge machine propulsor supports, and the number of kinematic coupling equations (11) is reduced by one due to the exclusion of angular rotation speed ω:

$$\omega = \frac{V_{ij}}{R_{ij}}. \tag{18}$$

The adequacy of partial models (17) was tested using the force and kinematic parameters of the bridge machine prototype movement (Fig. 2). Experimental verification of the main theoretical provisions was performed on the example of a stationary force (on-board) rotation of the prototype of a bridge vehicle, which allowed to exclude the influence of various random factors. The experiment consisted of 6 trials with different weights of the bridge machine with three times the number of repetitions of each trial. A simulation of the bridge machine loading was carried out by placing additional mass on it (Ostanin, 2022). During the experiment, the driving radius and torques on driving wheels were measured, with their subsequent recalculation to the moment of resistance to turning:

$$M_c=0.5\mathrm{B}\cdot(T_{y2}-T_{y1}). \tag{19}$$

The power diagram of the on-board turning prototype of the bridge machine is shown in Figure 7.

Following Figure 7, the turning model of a bridge vehicle consists of:

- three equations of motion (17):

$$\left.\begin{aligned} 0 &= \textstyle\sum_{i=1}^n\sum_{j=1}^m\left(T_{xij}\cdot\cos\gamma_{ij} - T_{yij}\cdot\sin\gamma_{ij} + f_{ij}\cdot G_{ij}\cdot\sin\gamma_{ij}\right) + P_x,\\ 0 &= \textstyle\sum_{i=1}^n\sum_{j=1}^m\left(T_{xij}\cdot\sin\gamma_{ij} + T_{yij}\cdot\cos\gamma_{ij} + f_{ij}\cdot G_{ij}\cdot\cos\gamma_{ij}\right) + P_y,\\ 0 &= \textstyle\sum_{i=1}^n\sum_{j=1}^m\left[M_{ij} + T_{yij}\cdot\sqrt{x_{cij}^2+y_{cij}^2} - f_{ij}\cdot G_{ij}\cdot\left(\sqrt{x_{cij}^2+y_{cij}^2}\right)\right] - M, \end{aligned}\right\} \tag{20}$$

- two equations of geometric relations (10):

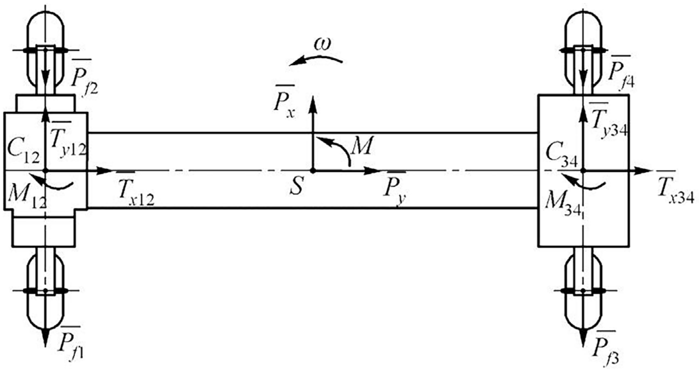

Figure 7. Diagram of forces acting on the bridge machine, when it performs a force (onboard) turn
Note: T_{xij}, T_{yij}, M_{ij} – force factors reduced to the centre of mass of the left and right sides of the bridge machine in contact *ij* of the support wheels with traces of a constant tramline; P_x, P_y, M – external forces and external moment reduced to the centre of mass of the machine; P_{fij} – resistance movement *ij* of the supporting wheel; ω – angular speed of rotation
Source: developed by the authors

$$\left.\begin{array}{r} y_0 + y_1 = 0, \\ y_1 = y_2, \end{array}\right\}, \tag{21}$$

- one equation of kinematic connections with one of the sides of the bridge machine disconnected:

$$T_{y1} = 0. \tag{22}$$

The adequacy of the model was evaluated by the force parameter – the coefficient of resistance to rotation:

$$\mu = \frac{4 \cdot M_c}{G \cdot L}. \tag{23}$$

The analysis of the convergence of theoretical and experimental values showed that the theoretical dependence $\mu(G)$ lies within the confidence interval $\pm\sigma$ throughout the investigated weight range G of the prototype bridge machine (Fig. 8) (Aghbalyan & Simonyan, 2022).

The obtained research results allow to assess the impact of design parameters and schemes on the characteristics of turning at the design stage of a new bridge machine, thereby optimizing its design. Issues related to the study of the traction qualities of wheeled vehicles are considered by A. Panchenko *et al*. (2019), and V. Bulgakov *et al*. (2021). They noted that for the machine as a whole, traction and dynamic properties are usually estimated by the traction efficiency and dynamic factor. However, each of the wheels of the bridge machine operates under certain conditions in terms of vertical load, input torque, and driving conditions. Therefore, the traction properties of the wheel of the bridge machine depend on a large number of parameters. These include its design parameters, on the one hand, and the physical and mechanical properties of the surface of permanent tramline tracks, on the other.

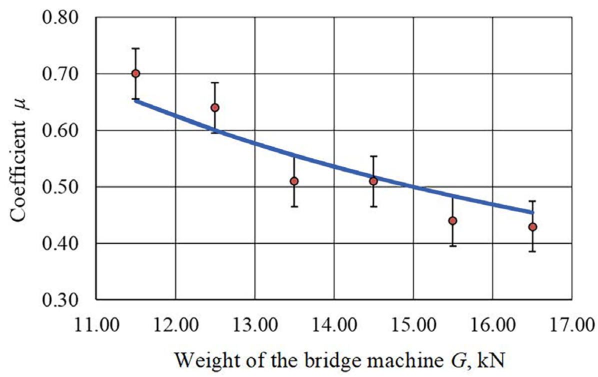

Figure 8. Theoretical dependence and experimental values of the coefficient μ from the weight G of the bridge machine
Note: μ – turning resistance coefficient; G – weight of the bridge machine
Source: developed by the authors

The obtained research results significantly supplement the known knowledge about the turning of wheeled vehicles, for example, presented by V. Nadykto *et al*. (2015) and S.P. Pozhidaev *et al*. (2016), who used the theory of active controlled movement (under the action of an operator) as a research basis. The obtained mathematical models consider the passive rotation of bridge machines in the process of their curvilinear movement without a control action from the operator. In particular, the uneven traction resistance that the bridge machine experiences along the width of its span; uneven vertical load on the support wheels from the left and right sides of the machine; the presence of support wheels for agricultural implements located in the area of the width of the span of the bridge machine and another, which redistributes its traction and coupling properties unevenly from its left and right sides. S.P. Pozhidaev *et al.* (2016) studied the kinematics and force interaction of the wheel or caterpillar of a traditional tractor with the soil at the moment of turning in detail. However, the issues of the curvilinear motion of a wide-gauge wheeled vehicle with a power (on-board) method of its control were not considered in this study, which makes it impossible to use the obtained theoretical and experimental studies to simulate the curvilinear motion of a bridge vehicle.

At the same time, the movement of the bridge machine along the soil tracks of a permanent tramline creates such conditions under which the modelling of forces in the contact of the mover with the surface of the tramline leads to the disappearance of the boundary between the wheeled and caterpillar mover and suggests the possibility of a unified approach to describing their force interaction with the support surface. Therefore, following M. Fashutdinov *et al.* (2020), greatly simplifies the process of studying the turning property of a bridge machine, even with a wheeled-caterpillar mover, as was considered, where the machine turn models are based on various methodological approaches when describing the interaction of the mover with the soil.

The obtained mathematical models of the curvilinear motion of bridge machines consider the variety of their design schemes and control systems, which impose certain restrictions on the characteristics of the curvilinear motion, manifested in the form of superimposed kinematic and geometric relationships. This significantly distinguishes the well-known theories of unsteady motion of a wheeled vehicle with front steered wheels, presented, for example, by V. Melnik *et al.* (2017).

T. Szakács (2010), I. Demšar *et al.* (2012), and M.Z. Song *et al*. (2014) researched large external loads acting on the bridge machine and the use of propulsion devices with small lugs on it was considered the presented studies to characterise their significant slip when turning within the boundaries of the permanent tramline track on the headland. Such an approach to describing the interaction of the propeller of a wide-span machine with the soil is absent when describing the movement of traditional mobile machines when they perform a turn.

Thus, the results of the study and its analysis showed that with an increase in the weight of the bridge machine due, for example, to an increase in its size or the use of heavier technological equipment on it, when moving along the soil track of a constant tramline, the resistance coefficient μ of its rotation tends to decrease. The obtained research results allow to address the problem of power rotation of a wide-span bridge machine, considering the slight deformation of the soil in the traces of a permanent tramline.

CONCLUSIONS

As a result of this research, a methodology for compiling private models of turning the bridge vehicle moving along the soil trace of the constant technological track was created. As a result of the joint solution problem of the bridge machine turning, it is possible to determine all output parameters of curvilinear motion: trajectory, tractive forces, turning radius, slipping, and actual speeds.

The force interaction of the bridge machine's undercarriage with the soil trace of a constant track is presented based on flat sliding with a variable anisotropic friction coefficient of adhesion φ_{yd}, depending on the properties of the track. As with the radius of the wheel, the width of its tire and the air pressure in it, as well as the vertical load that acts on it, the coefficient of traction of the bridge machine φ_{yd} increases, and more slippages can be achieved in doing so.

The adequacy of the model of stationary turning of the overhead machine is confirmed by experimental estimation of the resistance coefficient of the power on board turning. The convergence of the theoretical and experimental values of this coefficient is within the confidence interval $\pm\sigma$ in the whole investigated weight range of the prototype of the bridge machine. This allows to assess the influence of design parameters and schemes on the turning characteristics as early as the design stage of a new bridge machine, thus optimizing the design process.

The prospect for further research is the development of static and dynamic models of the turn of the bridge machine and, on their basis, to substantiate the parameters of the influence of structural and technological parameters on the dynamics of its turn, as well as the wear processes of its wheel tires.

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CONFLICT OF INTEREST

None.

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Аналітичне дослідження поворотів мостових машин

Анотація. Робота присвячена вирішенню актуальної проблеми ефективності повороту мостових машин у колійній системі землеробства. Метою роботи є дослідження криволінійного руху по слідах ґрунту постійної технологічної колії мостової машини з урахуванням її конструктивної схеми та способу повороту, параметрів, режимів руху та навантаження. Експериментальні дослідження проводились з використанням сучасної тензометричної доріжки та спеціально розробленого обладнання для електричних вимірювань неелектричних величин. Обробку даних дослідження проводили на персональному комп'ютері. В результаті проведених досліджень розроблено методику складання математичних моделей повороту мостової машини, яка рухається по ґрунтовому сліду постійної технологічної колії. Після спільного розв'язання моделей повороту мостової машини можна визначити всі вихідні параметри її криволінійного руху: траєкторію, сили тяги, радіус повороту, ковзання, фактичні швидкості. Представлено силову взаємодію ходової частини мостової машини з ґрунтовим слідом постійної колії на основі плоского ковзання зі змінним коефіцієнтом анізотропного тертя зчеплення φ_{yd} залежно від властивостей колії. Зі збільшенням радіуса колеса, ширини шини та тиску повітря в ньому, а також вертикального навантаження, що діє на нього, збільшується коефіцієнт зчеплення мостової машини φ_{yd}, при цьому можна досягти більшого пробуксовування. Адекватність моделі стаціонарного повороту підвісної машини підтверджено експериментальною оцінкою коефіцієнта опору силового бортового повороту. Збіжність теоретичних та експериментальних значень цього коефіцієнта знаходиться в межах довірчого інтервалу $\pm\sigma$ у всьому досліджуваному ваговому діапазоні прототипу мостової машини. Це дозволяє оцінити вплив конструктивних параметрів і схем на поворотні характеристики ще на етапі проектування нової мостової машини, таким чином оптимізуючи процес проектування. Результати досліджень можна застосувати до організації процесу повороту діючих моделей мостових машин

Ключові слова: кероване землеробство; портальні системи; криволінійний рух; коефіцієнт зчеплення; коефіцієнт адгезії