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Theoretical study of the movement of the wide span machine in quasi-static turning mode

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Abstract. Wide span machines represent sophisticated energy and technological tools for controlled traffic farming. The curvilinear movement (turning) of these machines are often decisive in the design of new or evaluation of existing models of equipment. The application of classical theory of turning in researching wide span machines faces certain challenges due to limitations imposed when describing the force or kinematic interaction of their movers with the supporting surface along a constant technological track. Additionally, non-traditional control schemes further complicate the use of the classical turning theory. The present research aims to study the curvilinear motion of wide span vehicles in a quasi-static turning mode, allowing for the modelling of their turns with combined or non-traditional movers under various control schemes. As a result of the conducted research, it was established that a promising non-traditional turning scheme for a wide span machine is one where one rear wheel is driving, and the other is braking, with controllable front wheels. In this case, the turning radius of the machine is 1.5 times smaller compared to traditional turning schemes. It was experimentally proven that the practical implementation of this new non-traditional turning scheme for the wide span machine is limited by the magnitude of the load on its front wheels relative to the rear ones.

Key words: controlled traffic farming, curvilinear motion, turning, wide span machine.

INTRODUCTION

Contemporary agricultural development trends, amidst heightened competition, necessitate the reduction of production costs per unit (McHugh et al., 2009; Tullberg, 2009; Bindi et al., 2013; Antille et al., 2015; Chamen, 2015; Antille et al., 2019). A pivotal focus of ongoing global research involves formulating a comprehensive set of technological measures to implement resource-efficient technologies and technical instruments, thereby diminishing the demand for material, technical, and energy resources. A prospective initiative in this context involves the implementation of wide span (gantry) systems in agriculture (Bochtis et al., 2010; Pedersen, 2011; Onal, 2012; Pedersen et al., 2016). The distinctive feature of these systems is the allocation of a portion of the overall field area for an engineering zone, encompassing a transportation system for the mobility of all mechanization equipment, communication infrastructure for energy and water supply, telecommunication channels, and an orientation system. The area of the engineering zone is primarily contingent upon the specifications of the transportation system, which, in a reciprocal manner, are dictated by the characteristics of technological and transport machinery. In controlled traffic farming systems, a promising innovation is a gantry machine modeled after the Dowler tractor. Within the operational cycle of any wide span machine, there exists a curvilinear motion (turning). The characteristics of this movement often play a decisive role in the design of new equipment or the assessment of existing models. Consequently, numerous scientists engage in the study of this specific type of motion (Rohde & Yule, 2003; Gasso et al., 2013; Gasso et al., 2014; Bulgakov et al., 2017; Bulgakov et al., 2019). Numerous scientific papers have been dedicated to the investigation of the turning of conventional machine-tractor aggregates (Gurudatta et al., 2018; Bulgakov et al., 2022), as the parameters of this type of motion largely determine its operational efficiency. Presently, two primary directions have emerged in the theory of turning, each employing distinct approaches to describing the motion. These directions encompass the theory of turning for wheeled and tracked machines. However, among contemporary tractor aggregates, machines featuring a combined or non-traditional type of mover are increasingly gaining prevalence (Ruiz-Garcia & Sanchez-Guerrero, 2022).

Attempts to apply existing models to describe the motion of such tractors (Hac et al., 2009; Adamchuk et al., 2016; Li et al., 2016; Bulgakov et al., 2018; Bulgakov et al., 2020) have generally been unsuccessful, primarily due to the constraints imposed when describing the force and kinematic interaction of their movers with the supporting surface. For instance, using the theory of turning for tracked vehicles (Ding & Tang, 2020) is deemed impractical for crawler tractors (such as those from Komatsu) or articulated-steering tractors (such as those from Case) due to inherent limitations in kinematics (non-turning supports of the mover).

To study the motion of conventional machine-tractor aggregates during turns, models based on the theory of lateral slip are most commonly employed (Nadykto et al., 2015). These models have found broad application in describing vehicles equipped with inter-axle differentials that operate virtually without wheel slip. Conversely, the classical theory of rotation for wheeled vehicles, based on the utilization of elastic side slip, precludes slipping relative to the ground. Simultaneously, non-traditional control schemes are increasingly being implemented in the models of experimental wide span machines. For example, additional braking during turns, accompanied by inevitable wheel slip (Bulgakov et al., 2019). In light of the aforementioned considerations, it is suggested to employ a methodology for simulating the curvilinear motion of wide span vehicles, conceptualized as a controlled entity whose curvilinear movement is governed by the constraints imposed on it, facilitated through the machine's design and its control system.

The aim of this study is to investigate the curvilinear motion of a wide span machine during a quasi-static turning mode, thereby enabling the simulation of its turning with a combined or non-traditional mover under various control schemes.

MATERIALS AND METHODS

An experimental research subject consisted of a prototype of a wide span machine (Fig. 1). The depicted wide span machine incorporates a power truss (1) functioning as a rigid frame connected to both the left (2) and right (3) platforms. Brackets (4) with wheels (5) are attached to platforms (2) and (3). The rotation of the wheels (5) is carried out using two gear motors (6) installed inside the platforms (2) and (3) and synchronous mechanisms (7) that drive the front and rear wheels (5) on each side of the machine. In the upper part of the left platform (2) there is a control cabinet (8) with chassis control devices. Supporting pillars (9) with power jacks for lifting the machine's sides from any direction are installed between the front and rear wheels (5) on the lower surface of platforms (2 and 3). Additional components of the machine's hydraulic system, including a hydraulic fluid reservoir (10), an electric motor (11) for driving the hydraulic pump (12), and a hydraulic distributor (13), are positioned on the crossbars of the frame (1). Additionally, a hydraulic suspended mechanism (14) is situated in the central part of the wide span machine.

The main technical specifications of the prototype (Fig. 1): operation mass 1,158 kg, wheel gauge 3.5 m, wheelbase 2.3 m, pneumatic tire size 9.5R32.

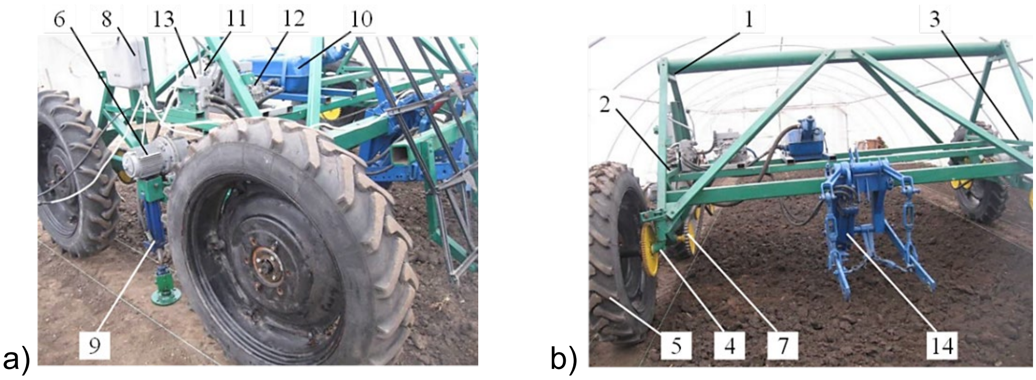

Figure 1. Overall view of the experimental wide span machine design: a) side view; b) front view: 1 – frame-truss; 2, 3 – left and right platforms; 4 – brackets; 5 – wheels; 6 – motor-reducers; 7 – wheel drive mechanism; 8 – control cabinet; 9 – support pillars with power jacks; 10 – hydraulic reservoir; 11 – electric motor; 12 – hydraulic pump; 13 – hydraulic distributor; 14 – suspended mechanism.

In the course of experimental investigations, the turning diameter of the machine was measured using a tape measure along the trajectory demarcated by inkjet markers affixed at the centre of its frame. The precision of the turning diameter measurement was within the range of ±1%.

The design of any agricultural gantry system permits two configurations: one with a wheeled mover and the other with a tracked mover. The wheeled mover configuration of the wide span machine continues to be the most promising. An evaluative analysis of the turning capability of the wide track vehicle equipped with a wheeled mover system facilitated the identification of the most rational area of application for each of its potential configurations (Fig. 2).

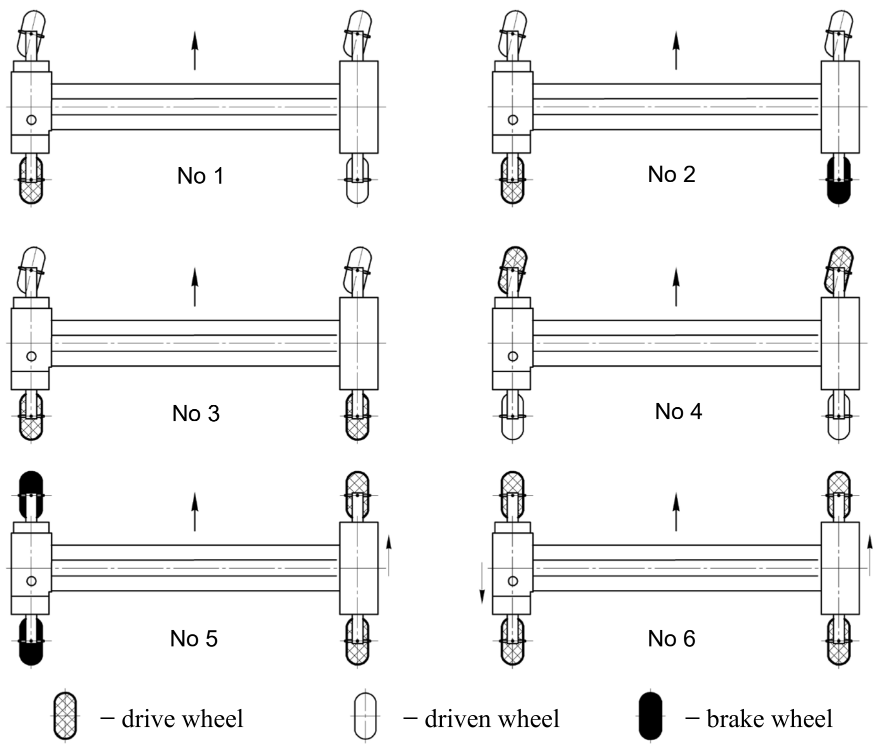

Figure 2. Schemes for controlling the turning of a wide span machine using one of its structural bases.

Initial investigations into the turning capacity of the wide-span machine indicate that Scheme No 1 is virtually unaffected by the vertical load on its drive wheels. Scheme No 2 exhibits the smallest turning radius among the configurations of the wide span machine. Nevertheless, when equipped with a front-mounted agricultural tool, resulting in an additional vertical load on the front of the machine, it forfeits its ability to move due to binding loss on the drive wheel. Schemes No 3 and No 4 are distinguished by their larger turning radius, yet they represent the most conventional and straightforward options for practical implementation. A wide span vehicle employing a skid steering turning, as demonstrated in Schemes No 5 and No 6, exhibits enhanced manoeuvrability. However, the utilization of a wheel propulsion device on a solid surface during turning results in increased slippage, leading to heightened wear on the vehicle wheels' tires.

In the course of the investigation, two non-traditional control schemes for the rotation of the wide span machine were examined: Scheme No 1 and No 2 (Fig. 2).

Theoretically, the analysed scheme of curvilinear motion of a physical wide span machine can be described in the mode of quasi-static turn (Fig. 3). This pertains to motion characterized by a variable radius, wherein inertial forces can be neglected and an equilibrium system of forces is present at each moment of time.

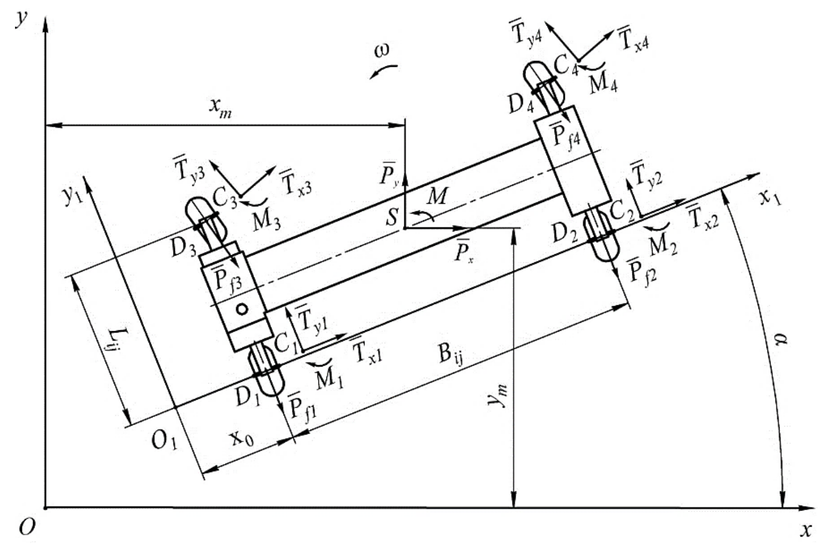

Figure 3. Scheme of quasi-static turning of a wide span machine with front swivel wheels.

The mathematical model of quasi-static turn of a wide span machine (Fig. 3) with *n* axes ($i = 1...n$) and *m* supports on each axis ($j = 1...m$) consists of 3 equations of motion:

$$\left.\begin{aligned}&\sum_{i=1}^{n}\sum_{j=1}^{m}F_{xij}=0,\\&\sum_{i=1}^{n}\sum_{j=1}^{m}F_{yij}=0,\\&\sum_{i=1}^{n}\sum_{j=1}^{m}M_{O_{ij}}=0,\end{aligned}\right\}\qquad(1)$$

where $\sum_{i=1}^{n}\sum_{j=1}^{m}F_{xij}$ – the sum of the projections of all forces on the *x* axis; $\sum_{i=1}^{n}\sum_{j=1}^{m}F_{yij}$ – the sum of the projections of all forces on the *y* axis; $\sum_{i=1}^{n}\sum_{j=1}^{m}M_{O_{ij}}$ – the sum of the moments of all forces about point *O* of each ij support of the wide span machine.

We write the equations of motion (1) of the wide span machine in the form:

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

where x_m, y_m, α – coordinates of the center of mass and the turning angle of the vehicle body in a fixed Cartesian system; P_{fij} – tractive resistance of ij supporting wheel; P_x, P_y, M – external forces and torque reduced to the center of mass of the machine; T_{xij}, T_{yij}, M_{ij} – force factors in contact of *ij* support wheel with the permanent tramline track; yij – turning angles of the *ij* support relative to the frame of wide span machine.

In non-stationary curvilinear motion, the steering wheels participate in two rotational movements about to parallel axes (Fig. 4). The first movement is the turning of the wheel together with the machine frame about the center of turning (point *O*), the second is a turning relative to the frame (point D_{ij}).

In this case, the constraint equations have the form:

$$\left(x_{cij} - x_{kij}\right) \cdot \sin\left(\alpha + \gamma_{ij}\right) - \left(y_{cij} - y_{kij}\right) \cdot \cos\left(\alpha + \gamma_{ij}\right) = 0, \tag{3}$$

$$\left(x_{cij} - x_{kij}\right) \cdot \sin\left(\alpha + \gamma_{ij}\right) - \left(y_{cij} - y_{kij}\right) \cdot \cos\left(\alpha + \gamma_{ij}\right) = 0, \tag{4}$$

where x_{kij}, y_{kij} – coordinates of point K_{ij} of ij support of wide span machine; x_{cij}, y_{cij} – coordinates of instantaneous center of zero velocity C_{ij} of support ij; α, γ_{ij}, $\dot{\alpha}$, $\dot{\gamma}_{ij}$ – turning angles and their time derivatives of the frame and ij support relative to the frame of wide span machine.

After transforming the coordinates and solving the equilibrium equations for the unknown coordinates of the instantaneous velocity center x_{ij}, y_{ij} of the wide span machine, we obtain:

$$x_{ij} = \frac{\left[\dot{x}_m \cdot \sin\left(\alpha + \gamma_{ij}\right) + V_{Tij} - \dot{y}_m \cdot \cos\left(\alpha + \gamma_{ij}\right) - \dot{\alpha} \cdot \left(B_{ij} \cdot \cos\gamma_{ij} + L_{ij} \cdot \sin\gamma_{ij}\right)\right]}{\dot{\alpha} + \dot{\gamma}_{ij}} \tag{5}$$

$$y_{ij} = \frac{\left[\dot{x}_m \cdot \cos\left(\alpha + \gamma_{ij}\right) + \dot{y}_m \cdot \sin\left(\alpha + \gamma_{ij}\right) + \dot{\alpha} \cdot \left(B_{ij} \cdot \sin\gamma_{ij} - L_{ij} \cdot \sin\gamma_{ij}\right)\right]}{\dot{\alpha} + \dot{\gamma}_{ij}}, \tag{6}$$

where B_{ij}, L_{ij} – transverse and longitudinal base of *ij* support of the wide span machine.

Mathematical models (5) and (6) represent equations of nonholonomic kinematic constraints for controlled curvilinear motion, reflecting the design parameters of the wide span machine and its control system.

The transformation of the coordinates of the instantaneous center of zero velocity (x_{cij}, y_{cij}) for any of its design of chassis is carried out by shift and turning. Equations of geometric constraints (Fig. 4), reflecting the construction arrangement and parameters of the wide span machine (base, track, number of supports and their relative positions) take the form:

$$\left.\begin{aligned} & y_0 + y_1 = 0, \\ & y_1 = y_2, \\ & x_{c3} \cdot \sin\gamma_3 = y_{c3} \cdot cos\,\gamma_3, \\ & x_{c4} \cdot \sin\gamma_4 = y_{c4} \cdot cos\,\gamma_4, \end{aligned}\right\} \tag{7}$$

where x_{c3}, x_{c4}, y_{c3}, y_{c4} – coordinates of the instantaneous center of zero velocity of support wheels 3 and 4 of the system of the wide span machine; y_{c3}, y_{c4} – turning angles of the support wheels 3 and 4 about the vertical axes in the system of the wide span machine.

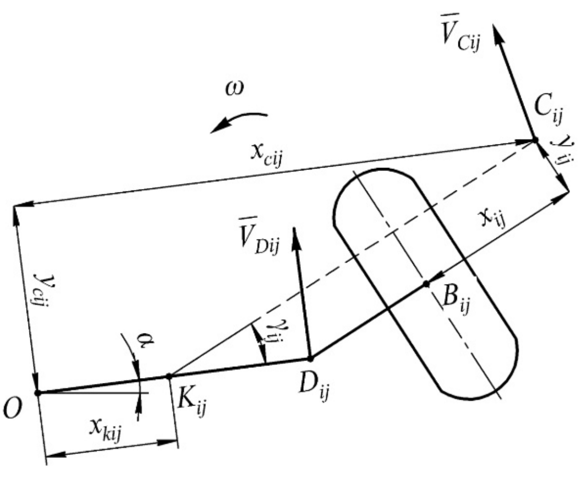

Figure 4. Kinematics of the steered wheel of a wide span machine.

The equations of kinematic constraints (4), depending on the implemented turning pattern of the wide span machine, reflecting the mode of motion of each support (driven, driving, brake) of the wide span machine, have the form:

$$\left.\begin{aligned} T_{y1} &= 0,\\ x_0 + x_1 &= 0,\\ T_{y3} &= 0,\\ T_{y4} &= 0. \end{aligned}\right\} \tag{8}$$

First order differential equations for constructing the turning trajectory of a wide span machine have the form:

$$\left.\begin{aligned} dx_m &= ds \cdot \cos\alpha,\\ dy_m &= ds \cdot \sin\alpha,\\ d\alpha &= \frac{ds}{\rho_m}, \end{aligned}\right\} \tag{9}$$

where $ds = V_m \cdot dt$ – differential of the trajectory arc of the center of mass of the wide span machine; V_m – theoretical velocity of the center of mass; ρ_m – radius of curvature of the trajectory of the center of mass.

To construct the trajectory of the center of mass of the wide span machine using the radius of curvature ρ_m, as a function of the time parameter τ, we will apply the theory of a curvilinear integral. Under zero initial conditions, this integral will take the following form:

$$x_m = \int_0^T V_m \cdot \cos\left(\int_0^t \frac{V_m}{\rho_m} d\tau\right) dt, \tag{10}$$

$$y_m = \int_0^T V_m \cdot \sin\left(\int_0^t \frac{V_m}{\rho_m} d\tau\right) dt, \tag{11}$$

where t, T – current time and process time, respectively.

The radius of curvature ρ_m of the trajectory of the center of mass of the wide span machine in this case is equal to

$$\rho_m = x_0 + x_m, \tag{12}$$

where x_0 – the initial coordinate of the trajectory curve of the center of mass of the wide span machine.

The information above indicates that constructing the trajectory of the center of mass of the wide span machine involves determining the values of x_0 and x_m in the moving coordinate system. The values of x_0 and x_m are easily calculated when solving the problem of initial motion, expressed in the form of equilibrium conditions for the wide span machine at different moments in time.

Therefore, theoretical modelling of the motion of the wide span machine in quasi-static turning mode allowed establishing a connection between the coordinates of the instantaneous centre of velocities of an arbitrarily located support of the mover and the centre of turning of the wide span machine. It enables the formulation of the missing constraint equations. Combining various geometric constraints (reflecting the structural scheme and parameters of the machine) with kinematic ones (reflecting the control system) allows describing all types of constraints imposed on the wide span machine during its curvilinear motion.

RESULTS AND DISCUSSION

The adequacy of theoretical models for the curvilinear motion of the wide span machine in quasi-static turning mode was assessed based on a kinematic parameter – the diameter of turning under different loads applied to its running gear.

The simulation of the load mode for the wide span machine was carried out by installing additional ballast located at its front part. The degree of loading on the front wheels of the wide span machine relative to the rear wheels varied in the range of $G_f/G_r = 0.5–2.5$. In the experiment, using traces of jet markers attached to the center of the wide span machine, the diameter of turning was measured.

The results of comparing the theoretical and experimental dependencies of the turning diameter of the wide span machine for control schemes No 1 and No 2 (Fig. 2) with the degree of loading on the front wheels of the wide span machine relative to the rear wheels are presented in Fig. 5 and Fig. 6, respectively.

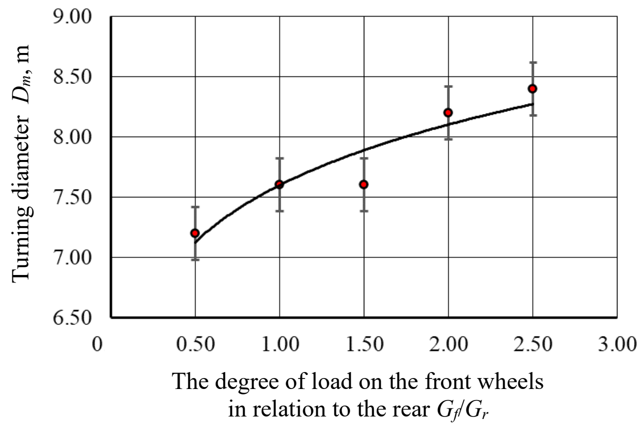

Figure 5. Comparison of theoretical and experimental dependences of the turning diameter of a bridge vehicle according to its control scheme No 1 (Fig. 2) on the degree of load on the front wheels in relation to the rear wheels.

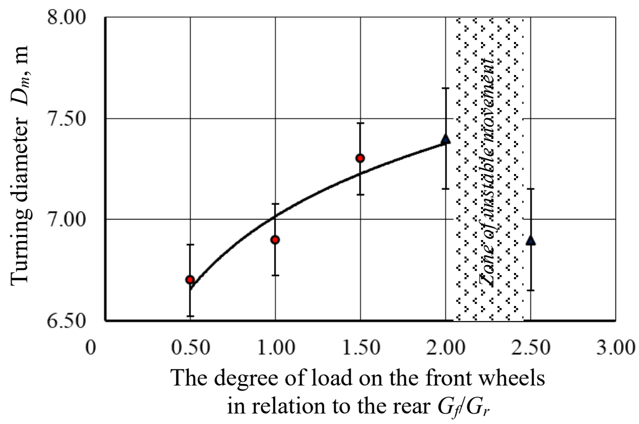

Figure 6. Comparison of theoretical and experimental dependences of the turning diameter of a bridge vehicle according to its control scheme No 2 (Fig. 2) on the degree of load on the front wheels in relation to the rear wheels.

The analysis of comparing the theoretical and experimental dependencies (Figs 5, 6) of the turning diameter of the wide span machine under different control schemes No 1 and No 2 (Fig. 2) with the degree of loading on its front wheels in relation to the rear ones showed that the maximum relative deviation of experimental points from the calculated curves across the entire load range $G_f/G_r = 0.5–2.5$ for both schemes did not exceed 11%.

It was experimentally established and theoretically confirmed that braking the rear inner wheel (scheme No 2, Fig. 2) leads to a 1.5-fold reduction in the turning radius. However, with an overload G_f/G_r of more than 2 times, the wide span machine loses its ability to move. During the experiments, this manifested as increased slipping, deviation from the stationary turning trajectory, and was characterized by a large spread in measurement results.

CONCLUSIONS

1. As a result of the conducted research, it has been established that a promising non-traditional turning scheme for the wide span machine is one where one rear wheel serves as the driving wheel, while the other functions as the braking wheel, with controllable front wheels. In this case, the turning radius of the wide span machine is 1.5 times smaller compared to traditional turning schemes.
2. When modelling the movement of the wide span machine in quasi-static turning mode, it is sufficient to have 3 equations of motion, 4 equations of geometric constraints, 3 equations of kinematic constraints, and 3 equations for constructing the turning trajectory. The adequacy of theoretical studies has been confirmed by experiments, where the deviation from theoretical dependencies does not exceed 11%.
3. It has been experimentally proven that the practical implementation of the mentioned new non-traditional turning scheme for the wide span machine is limited by the magnitude of the load on its front wheels in relation to the rear ones. With a load on the front wheels of the wide span machine more than 2 times that of the rear wheels, it loses the ability to move, resulting in increased slipping, and deviation from the stationary turning trajectory, and is characterized by a significant spread in measurement results.

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