STUDY OF INTER-WHEEL DIFFERENTIAL INFLUENCE ON DYNAMICS OF TRACTION AND TRANSPORT MACHINE

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Abstract. The results of studying the influence of the inter-wheel differential on the dynamics of a wheeled traction and transport machine (TTM) are presented. It has been established that there is a steady trend toward the development and operation of energy-saturated traction and transport machines (TTMs) that function as part of wide-field agricultural units. Locking the TTM differential has a positive effect on traction performance. In a locked drive, a fixed relationship exists between the angular velocities of TTM wheels. However, in such a system, parasitic power circulation within the transmission is possible. This leads to a decrease in the overall efficiency of TTM, increased tire wear, and higher dynamic loads on transmission components. To assess the influence of the inter-wheel differential on vehicle dynamics, a kinematic model of a traction and transport machine with an articulated frame was developed (on the example of the KhTZ-242K.20 tractor). The developed method for estimating the flow between the wheel differential on the trajectory of TTM half-frames and traction and energy indicators needs to be validated through experimental studies. During such studies, a measuring system of dynamics and energy of mobile machines was used. Additionally, the impact on the traction and energy characteristics of the machine was determined. The tangential traction forces on the wheels were measured, ranging from 3550 N to 4250 N for the front wheels and from 1325 N to 1855 N for the rear wheels of the all-wheel-drive TTM. The developed method for assessing the influence of the inter-wheel differential on the dynamics of a wheeled traction and transport machine should be considered valid.

Keywords: differential, articulation angles of half-frames, traction force, all-wheel drive wheeled traction and transport machine.

Introduction

A persistent trend is observed in the development and operation of high-energy traction-transport machines (TTM) that work as part of wide-coverage agricultural units. These agricultural units are to operate under increased speeds and loads, which means that TTMs should provide the highest traction performance. Differential locking in TTMs has a positive effect on traction performance. In a locked drive, there is a fixed ratio between the angular velocities of TTM wheels. In such a drive, parasitic power loss in the transmission is possible. This causes a decrease in the overall efficiency of the TTM, increased tire wear, and higher dynamic loads on the transmission components [1]. Experimental studies [2] have confirmed that locking the rear axle differential positively affects the stability of TTM movement on slopes. The development of new energy-efficient and high-power TTMs requires in-depth research into their dynamics and an assessment of the impact of locked and unlocked differentials. The issue of evaluating the influence of inter-wheel differentials on TTM dynamics remains underinvestigated. Increasing the directional stability of the machine during operation [3] is achieved by using a limited-slip differential. The blocking coefficient is controlled automatically, which improves stability. However, the dynamics of an all-wheel-drive machine with both front and rear locked differentials remain unexplored. The effect of the rear differential on both lateral and longitudinal stability has been studied [4]. The study determined wheel slip during acceleration in a front-wheel-drive electric vehicle and the angles of lateral deviation. However, due to the specific design features of the vehicle, the issue of evaluating energy performance under full differential locking has not been explored. The directional stability of vehicles in turns has been studied in [5], where it was found that using a controlled limited-slip differential improves stability at high speeds. However, the study did not examine the dynamics of a machine with both front and rear locked differentials and their impact on dynamic performance.

In works [7-9] vertical oscillations, stability of movement of the unit with a front-mounted implement and stability of movement of the modular unit are investigated. The works consider the movement of the tractor as part of the unit. The influence of the operating modes of the tractor differential on its dynamic and energy indicators has not been investigated. It should be noted that vertical oscillations significantly affect the dynamics of the tractor and the differential locking modes.

The review of the literature [1-9] has confirmed that differential locking in TTMs positively affects traction performance. However, the study of TTM dynamics with a locked differential remains insufficiently researched.

Materials and methods

When designing a traction and transport machine (using a machine-tractor unit with a wheeled tractor as an example), an essential task is evaluation of its dynamics. To assess the influence of the inter-wheel differential on the dynamics, a kinematic model of a traction and transport machine with an articulated frame has been developed (on the example of the KhTZ-242K.20 tractor) (Fig. 1).

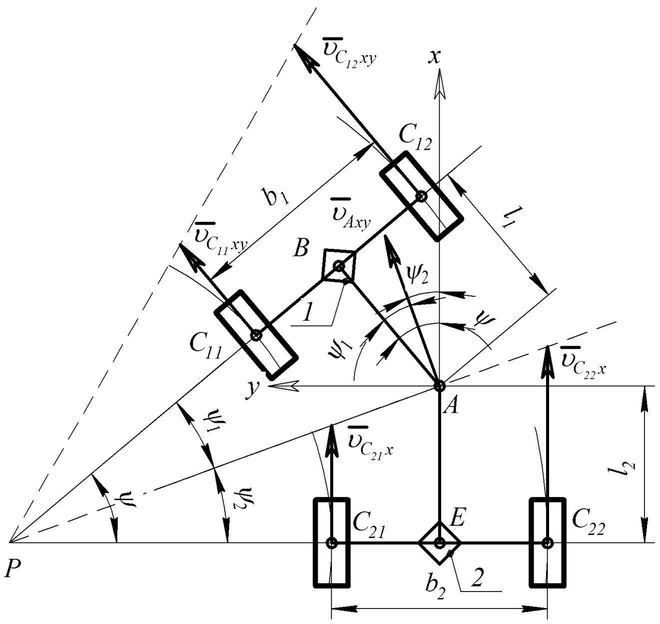


Fig. 1. **Calculated scheme of a tractor turn with an articulated frame:** 1 – front differential; 2 – rear differential; *point A* – center of turn of TTM half-frames; correspondingly, $\overline{v}_{Axy}$ – velocity component; horizontal velocity components of the wheel centers (in the plane of the TTM frame) – $\overline{v}_{C11xy}$, $\overline{v}_{C12xy}$, $\overline{v}_{C21xy}$, $\overline{v}_{C22xy}$; b_1, b_2 – track widths of the front and rear wheels; l_1, l_2 – distances from the hinge centre to the front and rear wheel axes, respectively; *points B* and *E* – centres of the front and rear axles; C_{11}, C_{12}, C_{21}, C_{22} – centres of the tractor wheels; instantaneous centre of velocities – *point P*; ψ – articulation angle of the TMM half-frames; ψ_1, ψ_2 – turning angles of the front and rear half-frames

The methodologies for evaluating the dynamics of wheeled vehicles are presented in studies [10; 11]. We will use the methodology for formulating the equations of motion for wheeled vehicles as described in studies [12; 13]. The velocities of the wheel centres in the horizontal plane are determined from the following equations:

$$v_{C_{i1}xy} = v_{Axy}\left(\cos\psi_i - \frac{b_i}{2l_i}\sin\psi_i\right),\ v_{C_{i2}xy} = v_{Axy}\left(\cos\psi_i + \frac{b_i}{2l_i}\sin\psi_i\right). \tag{1}$$

The equation (1) represents the equations of nonholonomic constraints, which ensure the rolling of the wheels within their respective planes.

The articulation angles of the TTM half-frames ψ_1, ψ_2 are determined from the following equations:

$$\psi_1 = \arcsin\frac{\sin\psi}{\sqrt{\sin^2\psi + \left(\frac{l_2}{l_1} + \cos\psi\right)^2}},\ \psi_2 = \arcsin\frac{\sin\psi}{\sqrt{\sin^2\psi + \left(\frac{l_1}{l_2} + \cos\psi\right)^2}}. \tag{2}$$

It should be mentioned that the equations (1) and (2) do not degenerate when $\psi = 0$.

The velocity projections of point A are determined from the equations:

$$\dot{x}_{A_i} = v_{A_i}\cos\left(\psi_1 + \psi\right),\ \dot{y}_{A_i} = v_{A_i}\sin\left(\psi_1 + \psi\right). \tag{3}$$

The results of solving the equations allow determining the orientation angles of the TTM half-frames (ψ_1, ψ_2) and their motion trajectories (x_{Ai}, y_{Ai}).

To assess the effect of the inter-wheel differential on the traction and energy performance of the traction and transport vehicle, a calculation scheme for the plane-parallel motion of the traction and transport vehicle was developed as part of the unit (Fig. 2).

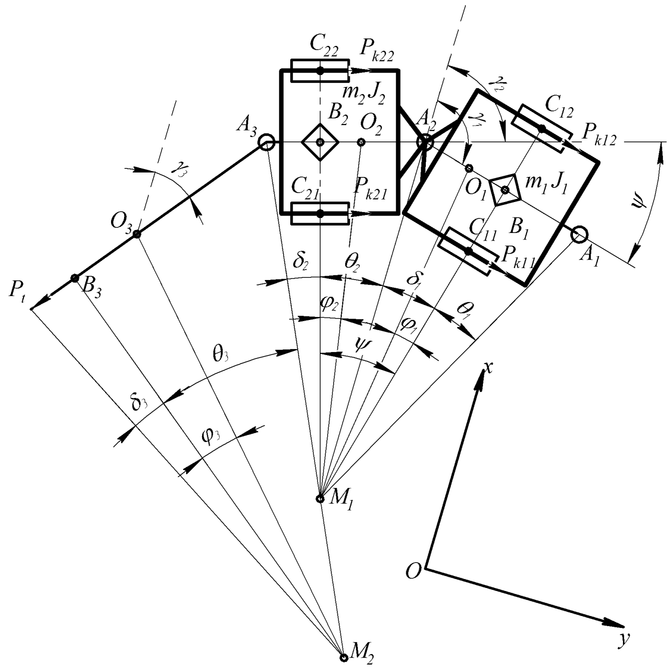


Fig. 2. **Calculated scheme of planar-parallel motion of the TTM:** C_{ij} – centres of the TTM wheels (i – row number, j – number in the row); A_k – front points of the frames (k = 1,2,3); O_k – centres of mass of the TTM half-frames; B_k – midpoint points of the half-frame axles; M_k – instantaneous centres of velocities (ICV) of the frame elements; ψ – articulation angle of the TTM half-frames; φ_k – angles between the lines of fixed bodies, wheel axes, and directions to the centres of mass from the relevant ICVs; θ_k – angles from the lines of the mentioned axes to the directions of the front hinges from ICVs; δ_k – angles from the axis lines to the directions of the rear hinges from ICVs; γ_k – angles between the longitudinal axes and the x-axis of the fixed coordinate system; m_1, J_1 – mass and moment of inertia of the first TTM half-frame relative to the vertical axis; P_{kij} – tangential traction forces on the TTM wheels; m_2, J_2 – mass and moment of inertia of the second TTM half-frame relative to the vertical axis; P_t – drawbar pull force on the TTM hook

As noted in the works [7-9], vertical vibrations have a significant impact on the dynamics of units. The dynamics of the tractor was studied exclusively as part of the unit. The TTM for which the scheme was drawn up (Fig. 2) was studied as part of the unit. The dynamics of an agricultural machine are described by the force P_1 applied to the tractor linkage mechanism at an angle γ_3.

The method for formulating kinematic constraint equations and dynamic equations using a specialized computer algebra system (SCAS ‘KiDiM’) is presented in works [12-14].

For the case of planar-parallel motion of TTM, considering the traction load represented by the force P_t, the dynamic equations are formulated as follows:

$$\begin{aligned} s\dot{A}_1 = {} & \frac{d\_sO_1}{\frac{dsA_1}{dt}}\cdot m_1\cdot s\ddot{O}_1 + \frac{d\_sO_2}{\frac{dsA_1}{dt}}\cdot m_2\cdot s\ddot{O}_2 + \frac{d\_\omega_1}{\frac{dsA_1}{dt}}\cdot J_1\cdot\dot{\omega}_1 + \frac{d\_\omega_2}{\frac{dsA_1}{dt}}\cdot J_2\cdot\dot{\omega}_2 + \\ & + - \frac{d\_sB_1}{\frac{dsA_1}{dt}}\cdot(P_{\mathrm{k}11}+P_{\mathrm{k}12}) - \frac{d\_sB_2}{\frac{dsA_1}{dt}}\cdot(P_{\mathrm{k}21}+P_{\mathrm{k}22}) - \frac{d\_sB_3}{\frac{dsA_1}{dt}}\cdot P_t \end{aligned} \tag{4}$$

and the system of kinematic constraints

$$\begin{cases} \delta_1 = \arcsin \dfrac{\sin\psi}{\sqrt{\sin^2\psi + \left(\dfrac{A_2B_2}{B_1A_2} + \cos\psi\right)^2}} ; \delta_n = \operatorname{arctg}\left(\lambda_n \cdot \operatorname{tg}\theta_n\right); \\ \theta_1 = \operatorname{arctg}\dfrac{\operatorname{tg}\delta_1 \cdot A_1B_1}{B_1A_2} ; \theta_2 = \arcsin \dfrac{\sin\psi}{\sqrt{\sin^2\psi + \left(\dfrac{B_1A_2}{A_2B_2} + \cos\psi\right)^2}} ; \\ \theta_n = \gamma_{n-1} - \gamma_n - \delta_{n-1} ; \phi_i = \operatorname{arctg}\left(\mu_i \operatorname{tg}\theta_i\right) ; \omega_i = s\dot{A}_i \dfrac{\sin\theta_i}{A_iB_i} ; \\ s\dot{A}_n = sA_n + s\dot{A}_{n-1}\sqrt{\left(\cos\theta_{n-1}\right)^2 + \left(\lambda_{n-1}\sin\theta_{n-1}\right)^2} ; \\ s\dot{B}_i = sB_i + s\dot{A}_i\cos\theta_i ; s\dot{O}_i = sO_i + s\dot{A}_i\sqrt{\left(\cos\theta_i\right)^2 + \left(\mu_i\sin\theta_i\right)^2} ; \\ \dot{\gamma}_i = \gamma_i + \omega_i ; x\dot{A}_i = xA_i + s\dot{A}_i\cos\left(\theta_i + \gamma_i\right) ; y\dot{A}_i = yA_i + s\dot{A}_i\sin\left(\theta_i + \gamma_i\right) ; \\ x\dot{B}_i = xB_i + s\dot{B}_i\cos\gamma_i ; y\dot{B}_i = yB_i + s\dot{B}_i\sin\gamma_i , \end{cases} \tag{5}$$

:

where s – pseudo-coordinates of element points;
x, y – coordinates of element points;
$i = 1,\ldots,3$ – element number representing the first half-frame of the TTM, the second half-frame of the TTM, and the equivalent tractive force;
$n = 2,\ldots,3$ – element number in the aggregate representing the second half-frame of the TTM and the equivalent tractive force.

The mathematical model of TTM dynamics, considering the influence of the inter-wheel differential (4) and (5), was solved in the SCAS 'KiDiM'. The initial data for calculating the mathematical model of the TTM are provided in works [14; 15].

The developed mathematical model of TTM dynamics, taking into account the influence of the inter-wheel differential, can be used to assess the impact of the differential on the dynamics of wheeled vehicles and agricultural tractors, both as part of machine-tractor units and separately.

The method developed for assessing the impact of the inter-wheel differential on the movement trajectories of TTM half-frames and traction-energy performance requires validation through experimental research. During such research, a measurement system for the dynamics and energy of mobile machines (MSDEMM) was used, its structural diagram is shown in Fig. 3.

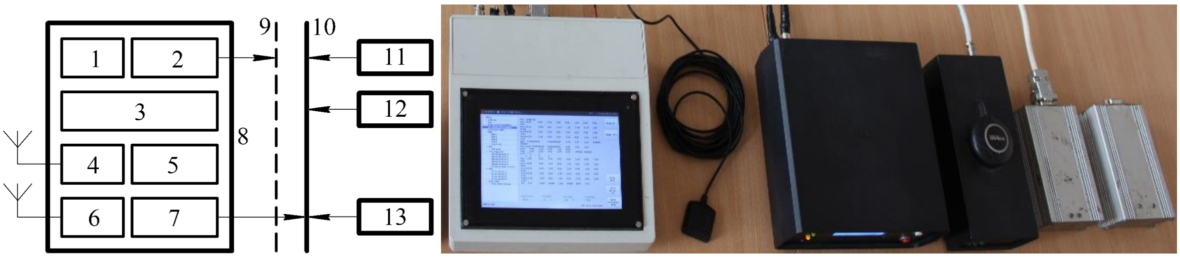


Fig. 3. **Structural diagram of the measurement system for dynamics and energy of mobile machines:** 1 – display; 2 – power supply; 3 – data storage (SSD); 4 – GPS receiver; 5 – Wi-Fi/Bluetooth radio module; 6 – 2.4 GHz radio module; 7 – computing module; 8 – main module of the measurement system; 9 – power line; 10 – CAN data bus; 11 – inertial measurement device 1; 12 – inertial measurement device 2; 13 – electronic dynamometer

The measurement system 8 includes two inertial measurement devices 11 and 12, which consist of a gyroscope, an accelerometer, and a magnetometer, as well as a navigation receiver 4 and an electronic dynamometer 13, which transmit data via the CAN bus to the system 8. For the measurement system, a software application DASys PC Suite has been developed, responsible for storing sensor data on an internal or external storage device.

Results and discussion

To assess the influence of the inter-wheel differential on the dynamics of TTM, theoretical studies were conducted for an all-wheel-drive articulated-frame wheeled tractor (KHTZ-242K.20). The dependencies of the front γ_1 and rear γ_2 half-frame steering angles around the vertical axis over time were determined as functions of the articulation angle ψ for both locked and unlocked differentials of the drive axles (Fig. 4). Additionally, the dependencies of the mass centre trajectories of the front (xA_1, yA_1) and rear (xA_2, yA_2) half-frames over time were determined as functions of the articulation angle ψ for locked and unlocked differentials of the drive axles (Fig. 5).

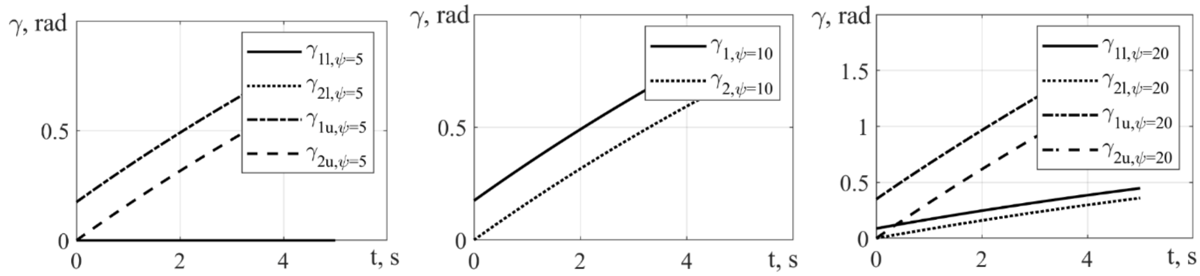


Fig. 4. **Dependencies of the front γ_1 and rear γ_2 tractor half-frame steering angles around the vertical axis over time (*l* – locked differentials, *u* – unlocked differentials) as a function of the tractor half-frame articulation angle ψ**

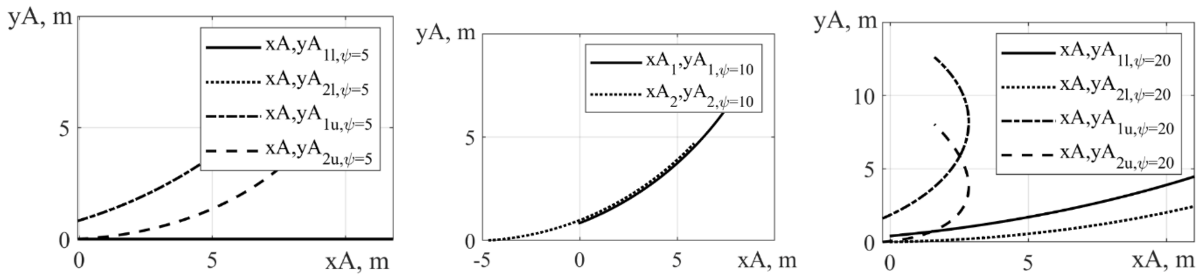
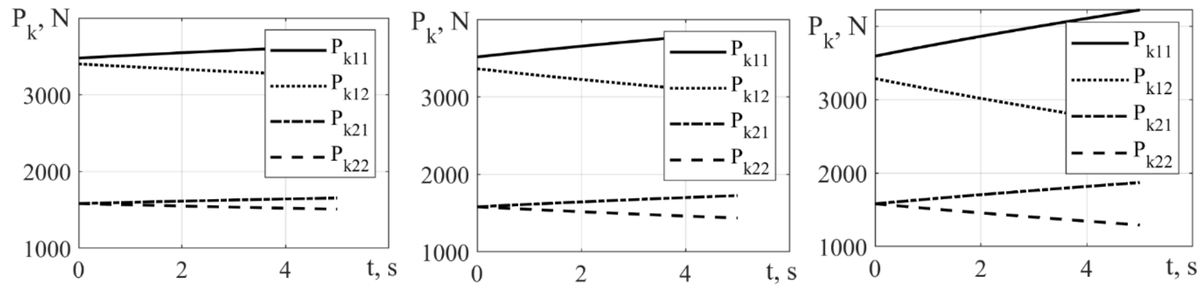


Fig. 5. **Dependencies of the mass centres' trajectories of the front (xA_1, yA_1) and rear (xA_2, yA_2) tractor half-frames over time (*l* – locked differentials, *u* – unlocked differentials) as a function of the articulation angle of the tractor half-frames ψ**

It has been determined that the movement trajectories of the tractor half-frames are linear when the differential is locked (Fig. 4). The steering angles of the tractor half-frames are $\gamma_1 = \gamma_2 = 0$ rad (when $\psi = 5°$ and $\psi = 10°$). When the differential is unlocked, the movement trajectories of the tractor half-frames are nonlinear. The steering angles of the tractor half-frames increase from $\gamma_1 = 0$ rad, $\gamma_2 = 0.05$ rad to $\gamma_1 = 0.89$ rad; $\gamma_2 = 0.71$ rad (when $\psi = 5°$); from $\gamma_1 = 0$ rad, $\gamma_2 = 0.01$ rad to $\gamma_1 = 1.78$ rad; $\gamma_2 = 1.434$ rad (when $\psi = 20°$).



$\psi = 5°$ $\psi = 10°$ $\psi = 20°$

Fig. 6. **Dependencies of the tangential traction forces on the wheels of TTM (front left P_{k11}, front right P_{k12}, rear left P_{k21}, and rear right P_{k22}) over time as a function of the tractor half-frame articulation angle ψ**

The dependencies of the tangential traction forces on the wheels of TTM over time as a function of the tractor half-frame articulation angle were determined (Fig. 6). It was established that the tangential traction forces on the front wheels of the all-wheel-drive TTM range from 3550 N ($\psi = 5°$) to 4250 N ($\psi = 20°$); for the rear wheels, the range is from 1325 N ($\psi = 5°$) to 1855 N ($\psi = 20°$). For the locked differential mode, the traction forces on the front wheels of TTM are 3530 N; on the rear wheels, they are 1870 N.

Experimental studies of the all-wheel drive wheeled TTM were carried out on the example of an all-wheel drive tractor with an articulated frame. The measurement of the functioning parameters was carried out by the measuring system of dynamics and energy of mobile machines. The methodology for conducting experimental studies using the measuring system of dynamics and energy of mobile machines is based on GOST 30745-2001 (ISO 789-9-90), GOST 7057-2001, GOST 24055-88 and is described in the works [13; 14].

During the experimental studies, the angles of rotation and the angle of break of the tractor half-frames, as well as the trajectories of their movement, were determined using the developed measuring system (Fig. 3). The angles of rotation of the tractor half-frames and the trajectories of movement were determined by inertial measuring devices consisting of an accelerometer, gyroscope and magnetometer. The accelerometer, gyroscope, and magnetometer signals are converted by the measuring system into motion trajectories and orientation angles of the unit's elements in space.

The results of theoretical and experimental studies were compared (Fig. 7).

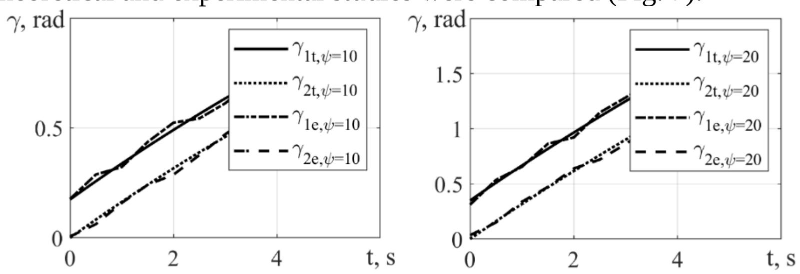


Fig. 7. **Dependences of angles of rotation of the TTM half-frames (obtained during theoretical studies – *t*, experimental studies – *e*) over time as a function of the tractor half-frame articulation angle ψ**

During experimental studies, using the developed measurement system (Fig. 3), the steering angles, the articulation angle of the tractor's half-frames, and their movement trajectories were determined. The results of theoretical and experimental studies were compared (Fig. 7). The discrepancy between the theoretical and experimental values of the steering angles, articulation angle of the half-frames, and movement trajectories of the TTM is 7%. Thus, the developed method for assessing the influence of the inter-wheel differential on the dynamics of a wheeled TTM should be considered valid. The method proposed in this study can be applied to evaluate the dynamics of wheeled motor vehicles.

To increase the traction performance of TTM as part of the agricultural unit, it is necessary to equip it with differentials with locking mechanisms. When the TTM is moving in a straight line, locking the differential leads to an increase in traction performance, namely, tangential traction forces on the wheels and a decrease in wheel slippage, especially when working on soils with poor bearing capacity. Differential locking should be performed at $\psi < 5°$. When turning ($\psi > 5°$), the differential must be unlocked. Differential locking should occur automatically depending on the angle of TTM half-frames. For this, it is recommended to equip TTM with an appropriate automatic differential locking control system.

Conclusions

1. It has been determined that the developed mathematical model of the TTM dynamics, taking into account the influence of the inter-wheel differential, can be applied to assess the impact of the differential on the dynamics of wheeled vehicles, agricultural tractors both as part of machine-tractor units and individually.
2. Theoretical studies have determined that the movement trajectories of the tractor half-frames are linear when the differential is locked. The steering angles of the tractor half-frames are $\gamma_1 = \gamma_2 = 0$

rad (when $\psi = 5°$ and $\psi = 10°$). When the differential is unlocked, the movement trajectories of the tractor half-frames are nonlinear. The steering angles of the tractor half-frames increase from $\gamma_1 = 0$ rad, $\gamma_2 = 0.05$ rad to $\gamma_1 = 0.89$ rad; $\gamma_2 = 0.71$ rad (when $\psi = 5°$); from $\gamma_1 = 0$ rad, $\gamma_2 = 0.01$ rad to $\gamma_1 = 1.78$ rad; $\gamma_2 = 1.434$ rad (when $\psi = 20°$). It was established that the tangential traction forces on the front wheels of the all-wheel-drive TTM range from 3550 N ($\psi = 5°$) to 4250 N ($\psi = 20°$); for the rear wheels, the range is from 1325 N ($\psi = 5°$) to 1855 N ($\psi = 20°$). For the locked differential mode, the traction forces on the front wheels of TTM are 3530 N; on the rear wheels, they are 1870 N.

3. During the experimental studies, the steering angles and the articulation angle of the tractor half-frames, as well as their movement trajectories, were determined, which allowed to compare the results of theoretical and experimental studies. The discrepancy between the theoretical and experimental values of the steering angles, articulation angle of the half-frames, and the movement trajectories of the TTM is 7%. It was established that the developed method for assessing the influence of the inter-wheel differential on the dynamics of the wheeled TTM should be considered valid.

Author contributions

Conceptualization, R. Antoshchenkov; methodology, I. Halych; software, A. Nikiforov; validation, R. Antoshchenkov and H. Cherevatenko; formal analysis, R. Antoshchenkov and S. Sheptun; investigation, A. Nikiforov; data curation, H. Cherevatenko; writing – original draft preparation, R. Antoshchenkov; writing – review and editing, I. Halych and A. Nikiforov; visualization, S. Sheptun. All authors have read and agreed to the published version of the manuscript.

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