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INVESTIGATIONS OF THE DYNAMICS OF A FOUR-ELEMENT MACHINE-AND-TRACTOR AGGREGATE

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The paper presents a dynamic model of four-element machine-and-tractor designed on the basis of a combined soil-cultivating-sowing aggregate and a tractor with an articulated frame. The pseudo velocity of the first tractor half-frame was considered a generalised coordinate. The control impact upon the tractor was obtained by means of the bending angle of the tractor half-frames. The trajectories and movement speeds of the aggregate elements were calculated as a function of time for the driving mode along a straight-line path and the tractor movement according to the harmonic control law. By means of the measuring system, trajectories of the aggregate elements were obtained. It was established that the movement trajectories of the aggregate elements represent a complex assessment parameter of the mathematical model adequacy of the aggregate plane-parallel movement. There was found a regularity of the aggregate turning radius and the bending angle of the tractor half-frames.

Keywords: dynamics; movement; mathematical mode; measuring system

Increased crop production efficiency can be achieved by using combined tillage-sowing aggregates, which can perform two or more technological operations in one field pass. Such aggregates are usually multi-element machines consisting of three elements – a tractor, a seed tank and a sowing machine – moving in succession one after the other (Beloev et al., 2015; Ivanovs et al., 2018). The most frequent layout schemes of such aggregates are: 1 tractor – tank for seed material – sowing machine; 2 tractor – sowing machine – tank. Differences in the dynamics of the aggregates with either of the two schemes are insufficiently studied.

Conventional ways of constructing dynamic and mathematical models for such systems are based on a manual technology (Blundell and Harty, 2004; Werner, 2012; Fleischmann and Berns, 2013). Recently, systems of a computer output of the dynamics equations have been paid much attention. A quite efficient one is the special system of computer algebra SCCA KiDiM, the theoretical foundations of which were described by Franke et al. (2012). The studies of the dynamics of the multi-element machine-and-tractor aggregate (MTA) are traditionally based on the formation of holonomic equations and nonholonomic systems (Blundell and Harty, 2004). Taking into account the regularities of rolling torque of the aggregate wheels, including even their elasticity properties, it belongs to the class of nonholonomic mechanical systems. In order to investigate the dynamics of multi-element machines, the D'Alembert-Lagrange principle (Bulgakov et al., 2016) or the Lagrange equations of the 2nd kind can be applied. Usually, research of the mobile machine movement is studied together with a semi-trailer by means of the Lagrange equations of the 2nd kind (Chieh, 1995; Siew et al., 2009; Bulgakov et al., 2018). A mathematical model of the single machine movement has been repeatedly investigated (Blundell and Harty, 2004; Karkee and Seward, 2010). Liljedahl et al. (1996) investigated the dynamics and stability of a mobile machine. Aforementioned works examine the mathematical model of the multi-element aggregate movement as integral; therefore, with a change in the structure or internal connections, it is necessary to reconstruct it again, resulting in more labour and time necessary for the research. Therefore, in order to correctly solve the problems of the dynamics of nonholonomic multi-element systems, it is necessary to establish the basic equations of dynamics and substantiate the equations of relations similarly to Franke et al. (2012). Numerous studies on the dynamics of agricultural aggregates as multi-element (multi-mass) mechanical systems do not try to deal with the scientific issue by an arbitrary combination of system elements. Investigation of the dynamics of these systems requires a methodology development for the formation of dynamics equations taking into account the changes in the structures and methods of machine aggregation. Methodology for establishing the equations of the dynamics of MTA cannot be used in study of aggregates with interchangeable structures, since it does not allow changing the mathematical model depending on the aggregation scheme of the multi-element aggregate elements. Known measuring systems of the MTA dynamics are either universal or utilized in automated control systems of the technological

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processes. A disadvantage of such systems lies in possibility to control only the intermediate values of the aggregate performance parameters with their help. Other systems utilize a small number of sensors and therefore can measure only a limited number of aggregate performance parameters (Yahya et al., 2004). The trajectories of a tractor or car movement are determined using the GPS receivers installed on machines (Yahya, 2000).

Investigation presented aims to study theoretically the dynamics of a multi-element MTA and to compare it with experimental data through mathematical simulation of a dynamic model based on the HTZ-17224 tractor and a combined soil-cultivating-sowing aggregate APP-6.

Material and methods

In order to conduct the investigation, methods of higher mathematics and theoretical mechanics were used for the solution of systems of nonlinear differential equations; bringing the dynamic equations to the Cauchy form was carried out by the Kratu method; numerical integration of such equations was carried out by Runge-Kutta methods with automated step selection, as well as the methods of programming and experimental field research. Processing of the results of both theoretical and experimental investigations was carried out by numerical methods using a PC. To compile an analytical mathematical model of a multi-element MTA, a design scheme (Fig. 1) will be used. The tractor with an articulated frame aggregates a tank for seed material and a sowing machine. When creating a dynamic model, it is necessary to take into account only the main elements of the multi-element MTA that affect its dynamics.

To construct an equivalent dynamic model, the following notation will be used (Fig. 1): C_{ij} – wheel centres (i – row number, j – number in the row); A_k – frame frontal points (k = 1, 2, 3, 4, for a four-element aggregate); O_k – mass centres of tractor, seed tank and sowing machine; B_k – rear axle midpoint of the tractor, seed tank and sowing machine; P_k – instantaneous speed centres (ISC) of the aggregate frame elements; Ψ – bending angle of the tractor half-frames; Φ_k – angles between the lines, fixed bodies, wheel axles, and directions to the mass centres of the corresponding ISC; θ_k – angles from the lines of indicated axles to the directions of the frontal joints of the ISC; δ_1, δ_2 – angles from the axle lines to the directions of the rear joints from the ISC; γ_k – angles, longitudinal axles of the vehicles with axis x of the fixed coordinate system; F_{Bk} – forces acting upon the aggregate elements. This system of differential equations consists of equations of kinematic links and an equation of dynamics. The equations of kinematic links describe links between the projections of the linear and angular velocities, and accelerations. For a four-element MTA, the equations of kinematic links are as follows:

$$\delta_1 = \arcsin\frac{\sin\psi}{\sqrt{\sin^2\psi + \left(\frac{A_2B_2}{B_1A_2} + \cos\psi\right)^2}}$$

$$\delta_n = \arctan(\lambda_n \cdot \tan\theta_n)$$

$$\theta_1 = \arcsin\frac{\sin\psi}{\sqrt{\sin^2\psi + \left(\frac{B_1A_2}{A_2B_2} + \cos\psi\right)^2}}$$

$$\theta_n = \gamma_{n-1} - \gamma_n - \delta_{n-1}$$

$$\varphi_i = \arctan(\mu_i \tan\theta_i)$$

$$\omega_i = s\dot{A}_i \frac{\sin\theta_i}{A_iB_i} \qquad (1)$$

$$s\dot{A}_n = sA_n + s\dot{A}_{n-1}\sqrt{(\cos\theta_{n-1})^2 + (\lambda_{n-1}\sin\theta_{n-1})^2}$$

$$s\dot{B}_i = sB_i + s\dot{A}_i\cos\theta_i$$

$$s\dot{O}_n = sO_i + s\dot{A}_i\sqrt{(\cos\theta_i)^2 + (\mu_i\sin\theta_i)^2}$$

$$\dot{\gamma}_i = \gamma_i + \omega_i$$

$$x\dot{A}_i = xA_i + s\dot{A}_i\cos(\theta_i + \gamma_i)$$

$$y\dot{A}_i = yA_i + s\dot{A}_i\sin(\theta_i + \gamma_i)$$

$$x\dot{B}_i = xB_i + s\dot{B}_i\cos\gamma_i$$

$$y\dot{B}_i = yB_i + s\dot{B}_i\sin\gamma_i$$

where:

- s – pseudo-coordinates of the points of aggregate elements
- x, y – coordinates of the points of aggregate elements
- i = 1…, 4 – number of the aggregate element, designating the first half-frame of the tractor, the second half-frame of the tractor, the seed tank, and the sowing machine, respectively
- n = 2…, 4 – number of the aggregate element similarly denoting the second half-frame of the tractor, the seed tank, and the sowing machine

Pseudo-velocity of point A – $s\dot{A}_1$ is taken as the generalised coordinate. The mathematical model has one degree of freedom, pseudo-velocity of point A, and 29 coordinate equations

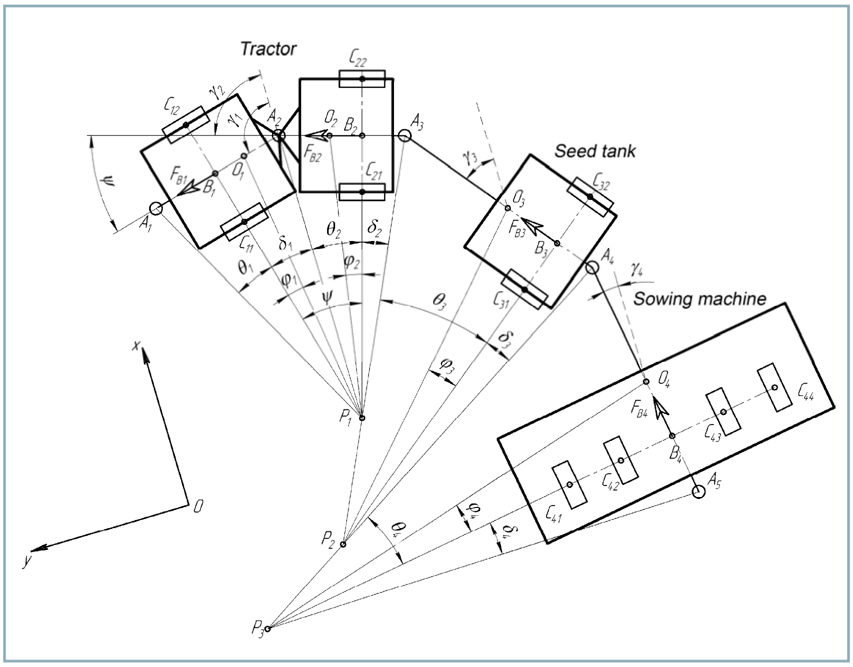

Fig. 1 Design scheme of a multi-element MTA

with dependent variations (Eq. 1). The dynamic equation is as follows:

$$s\dot{A}_1 = M_f[1][1]^{-1}\cdot F_f[1] \tag{2}$$

where:

factors $M_f[1][1]^{-1}$ and $F_f[1]$ of Eq. 2 are:

$$M_f[1][1]^{-1} = m_1(\cos^2\theta_1 + 0.12755\sin^2\theta_1) + m_2\left(\frac{dsO_2}{\frac{dsA_1}{dt}}\right)^2 + m_3\left(\frac{dsO_3}{\frac{dsA_1}{dt}}\right)^2 + m_4\left(\frac{dsO_4}{\frac{dsA_1}{dt}}\right)^2 + J_1\left(\frac{d\omega_1}{\frac{dsA_1}{dt}}\right)^2 + J_2\left(\frac{d\omega_2}{\frac{dsA_1}{dt}}\right)^2 + J_3\left(\frac{d\omega_3}{\frac{dsA_1}{dt}}\right)^2 + J_4\left(\frac{d\omega_4}{\frac{dsA_1}{dt}}\right)^2 \tag{3}$$

$$F_f[1] = -m_1\frac{dsO_1}{\frac{dsA_1}{dt}}\cdot s\ddot{O}_1 - m_2\frac{dsO_2}{\frac{dsA_1}{dt}}\cdot s\ddot{O}_2 - m_3\frac{dsO_3}{\frac{dsA_1}{dt}}\cdot s\ddot{O}_3 - m_4\frac{dsO_3}{\frac{dsA_1}{dt}}\cdot s\ddot{O}_4 - J_3\frac{d\omega_1}{\frac{dsA_1}{dt}}\cdot\ddot{\omega}_3 + J_4\frac{d\omega_4}{\frac{dsA_1}{dt}}\cdot\ddot{\omega}_4 - \frac{dsB_1}{\frac{dsA_1}{dt}}\cdot F_{B_1} + \frac{dsB_2}{\frac{dsA_1}{dt}}\cdot F_{B_1} + \frac{dsB_2}{\frac{dsA_1}{dt}}\cdot F_{B_3} - \frac{dsB_4}{\frac{dsA_1}{dt}}\cdot F_{B_4} \tag{4}$$

For the purpose of experiments, a measuring system was developed in order to determine the dynamic and traction-energy indicators of mobile machines. The primary element of this system is the computation module, since processing and storing of data obtained from sensors and measuring devices take place in this unit. A hard disk or a USB flash drive was used as a data storage device. The computation module includes a touch screen display for showing the recorded parameters and controlling the measuring system, making it usable without a PC. Block diagram of the measuring system for determination of dynamic and traction-energy indicators of the performance of mobile machines is presented in Fig. 2.

The system can process data by eight inertial measuring devices (IMD) [6]. These devices are used to determine the accelerations, angular velocities, field directions and the Euler angles. Furthermore, wheel dynamics sensors [14] also contain inertial measuring devices. They are installed in the wheel turning centre and transmit data via a 2.4 GHz radio channel. Such sensors allow the measuring of the slipping of tractor wheels by the following equation:

$$\delta = \frac{2\pi\cdot\omega\cdot r_D - \nu_D}{\nu_D}\cdot 100\% \tag{5}$$

where:

ω – wheel current turning speed determined by the sensor
r_D – real radius
υ_D – actual driving speed measured by the GPS receiver

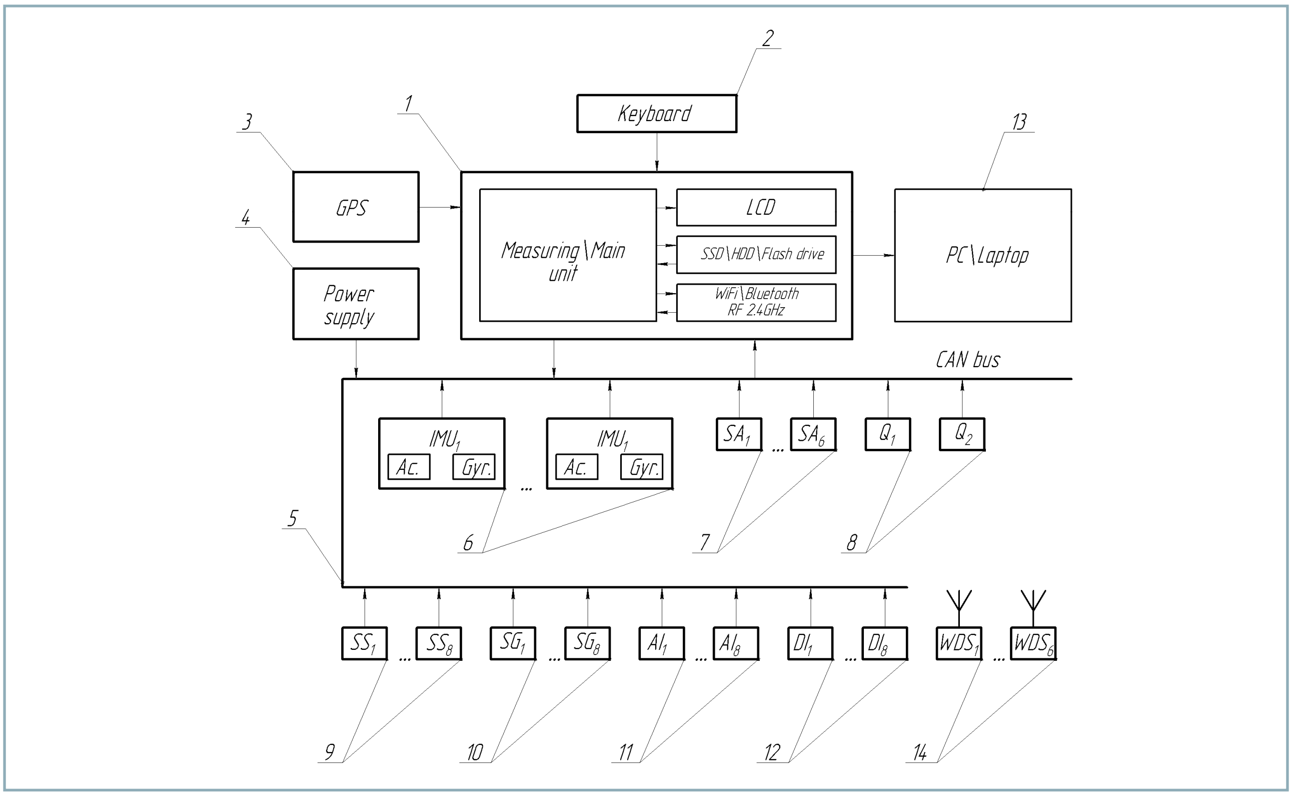

Fig. 2 Block diagram of the measuring system for determination of dynamic and traction-energy indicators of the performance of mobile machines
1 – computation module; 2 – control panel; 3 – GPS receiver; 4 – power supply; 5 – CAN data bus; 6 – inertial measuring device; 7 – turning angle sensor; 8 – fuel flow meter; 9 – revolution speed sensor; 10 – electronic dynamometer; 11 – analogue inputs; 12 –discrete inputs; 13 – PC; 14 – wheel dynamics sensor

This method of determining the turning speed of the tractor wheels allows a dynamic exploration of the slip for each of the wheels at a particular moment of time (the instantaneous value). The method of measuring the wheel turning speed helps to prevent the need for utilization of expensive precision encoders, as well as for changes in the tractor transmission design in order to install the turning speed sensors based on the Hall effect.

Results and discussion

Solving of the system of differential equations (Eqs. 1–4) was performed in a special system of the computer algebra SCCA KiDiM. The HTZ-17224 tractor and the APP-6 – a combined soil-cultivating-sowing aggregate – were selected as objects of theoretical studies. Control of the tractor is exerted by the bending angle of the tractor half-frame ψ. The theoretical research results are presented in Figs. 3–5.

The aggregate rectilinear movement scheme (Fig. 3) was obtained under the condition that the bending angle of the tractor half-frames was equal to zero $\psi = 0°$. The actual movement trajectory of the aggregate elements varies according to the harmonic law (Denham et al., 2007). Tractor with an articulated frame changes the trajectory of movement by the bending angle of semi-frames given by the law $\psi = 0.2 \sin(0.5t)$. When moving along a sinusoidal path, the bending angles of the first and the second tractor half-frames coincide (Fig. 4), the range of their oscillations is $\Delta\gamma_1 = \gamma_2 = 0.77$ rad, and the period is $T = 12.5$ s. The oscillation range of the angles (γ) of the seed tank and sowing machine is $\Delta\gamma_3 = 0.67$ rad and $\Delta\gamma_4 = 0.45$ rad with a period $T = 12.5$ s, which indicates a smaller deviation of these elements from the rectilinear line movement path.

The movement velocities of the mass centres of aggregate elements when moving along a sinusoidal path are shown in Fig. 5. A decrease in these velocities was observed at the movement beginning at $t \leq 7.5$ s. The seed tank mass centre showed the lowest velocity at the beginning of the movement $s\dot{O}_3 = 2.75$ m·s^{-1}, with an oscillation range of 0.12 m·s^{-1} and a period $T = 12.5$ s. The velocity oscillation range of the mass centres of the first and second aggregate half-frames was 0.08 m·s^{-1}; this oscillation range was 0.02 m·s^{-1} for the sowing machine. In order to evaluate the obtained theoretical model, a comparison of the trajectory of the mass centres of aggregate elements was experimentally carried out (Fig. 6).

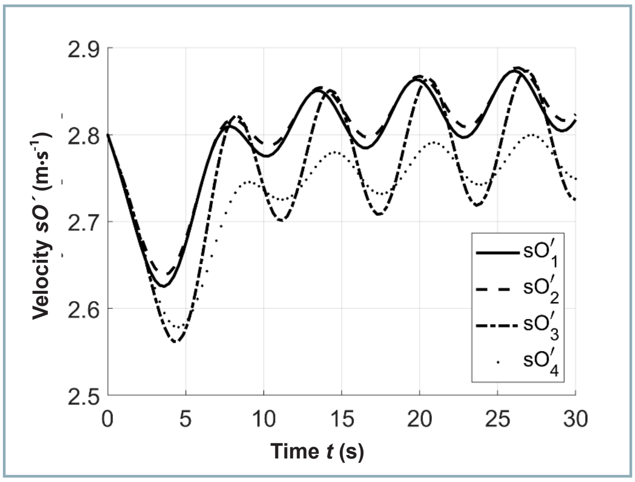

Fig. 4 Dependence of the turning angles of aggregate elements on time

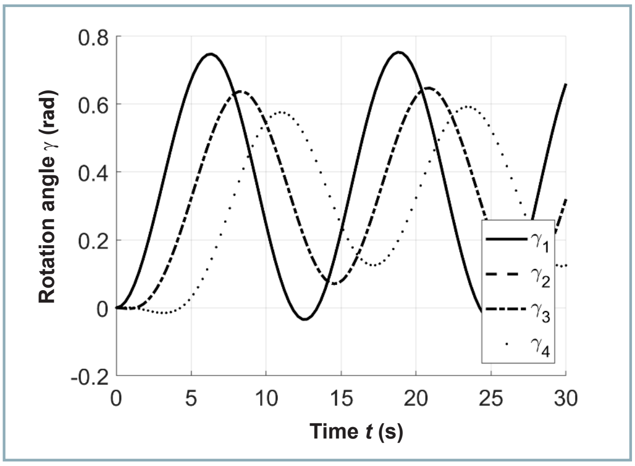

Fig. 5 Dependence of the velocities of the mass centres of aggregate elements on time when moving along a sinusoidal path

At a fixed bending angle of the tractor half-frames $\psi = 10°$, the aggregate turning radius is $r_p = 16.25$ m. Obtained as a result of experimental and theoretical research, the discrepancy between the values of the trajectories of the tractor, seed tank and sowing machine does not exceed 4%. The trajectories of the aggregate elements represent a complex parameter for the assessment of the mathematical model adequacy of an aggregate plane-parallel movement.

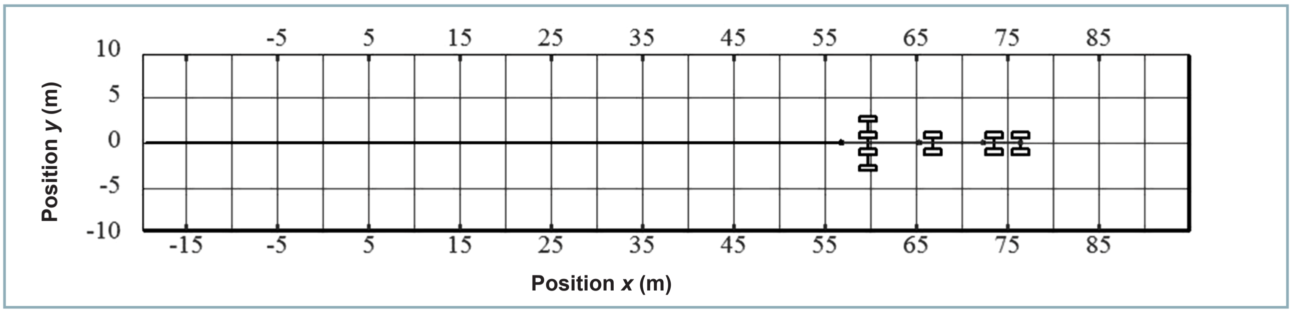

Fig. 3 Scheme of the aggregate rectilinear movement

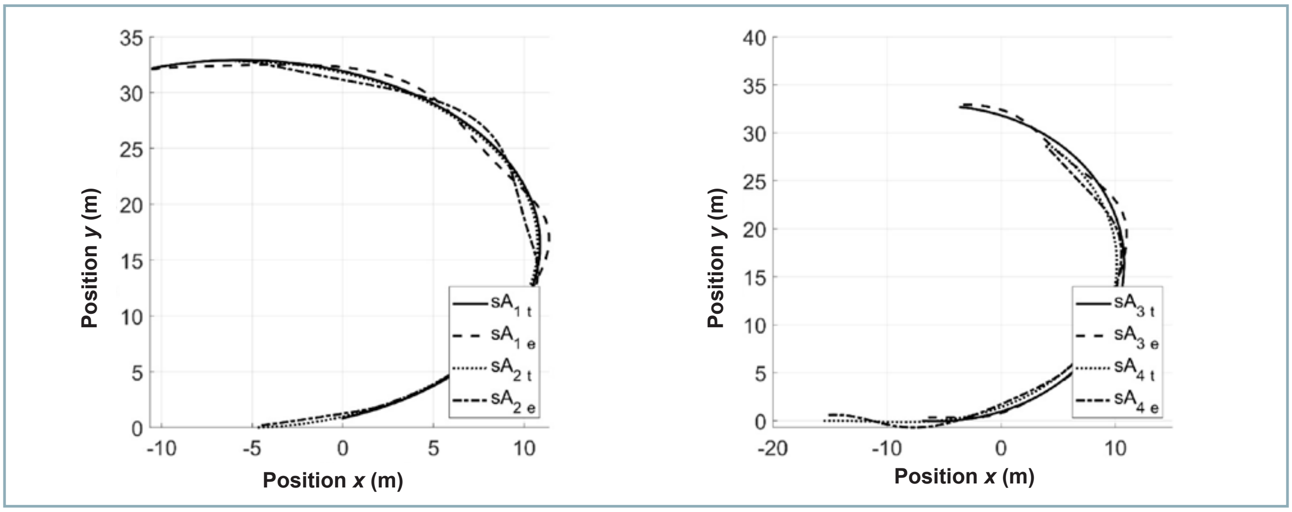

Fig. 6 Comparison of the movement trajectories of aggregate elements HTZ-17021 + APP-6, obtained from theoretical and experimental studies (the trajectories of the first and second tractor half-frames at experimental sA_{1e}, sA_2e, sA_3e, sA_{4e}; seed tank and sowing machine at theoretical sA_{1e}, sA_{2e}, sA_{3e}, sA_{4e}

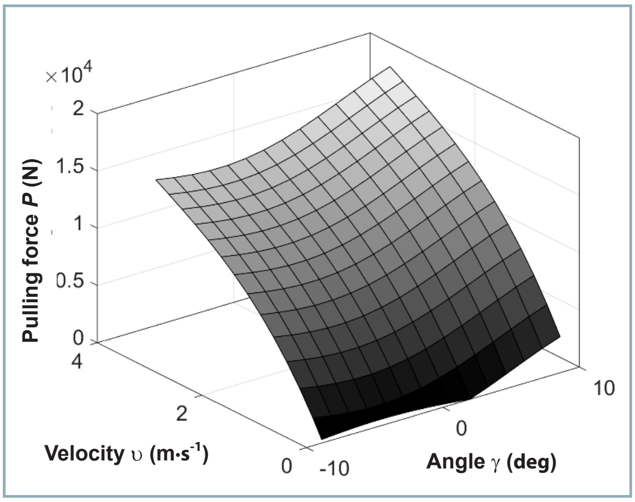

Fig. 7 Draft resistance dependence of the sowing machine APP-6 on the turning angle and its movement speed

By means of the GNU Octave software, draft resistance dependence of the APP-6 sowing machine on the movement speed and the turning angle was determined as follows:

$$P(\gamma, \upsilon) = -486.8 + 279.2 \cdot \gamma + 8{,}100 \cdot \upsilon + 46.03 \cdot \gamma^2 - 3.2 \cdot 10^{-14} \cdot \gamma \cdot \upsilon - 1{,}018 \cdot \upsilon \quad \text{(N)} \qquad (6)$$

where:

γ – sowing machine turning angle around axis z (°)

υ – sowing machine movement speed (m·s^{-1})

The determination coefficient for polynomial (Eq. 6) is R^2 = 0.9532, which indicates the dependence and its correctness. The sowing machine draft resistance value for its movement speeds ranging from 0 to 3.25 m·s^{-1}, and the turning angle around axis *z* within the limits $-10° \leq \gamma \leq 10°$ is shown in Fig. 7.

The sensors of the tractor wheel dynamics allow detection of their slipping. For a wheeled tractor, the slip should not exceed 15% if energy-intensive measures were carried out. Let us compare the theoretical υ_T and the actual

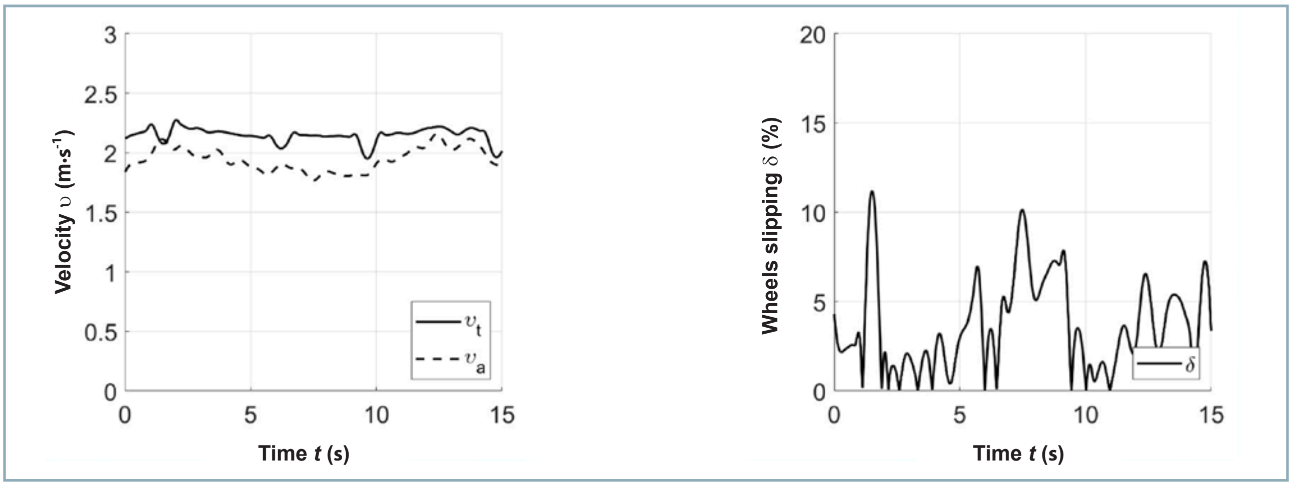

Fig. 8 Dependences of the aggregate actual speed (a) and the slipping of the tractor propulsors (b) on time

speed υ_D of the aggregate while sowing the grain crops (Fig. 8a) and calculate the slip of the tractor propulsors δ (Fig. 8b). The average aggregate actual speed value is $\delta_D = 2$ m·s^{-1}. The average slipping value $\delta = 5\%$; the maximum slipping value is $\delta_{max} = 11.5\%$, which meets the agrotechnical requirements.

On the basis of the obtained dynamic parameters of the HTZ-17021 tractor and the APP-6 sowing machine, it is clear that it works under satisfactory conditions with slipping $\delta = 5\%$; and with its maximum value $\delta_{max} = 11.5\%$, the turning radius $r_p = 16.25$ m at the end of the pass, which is less than its kinematic length $l_k = 18$ m.

The solution of the system of equations, which describes the dynamics of a multi-element MTA, made it possible to obtain dependences of the rotation angles of aggregate elements on time. This solution takes into account the tractor half-frame breakage angle ψ. The turning angles of the aggregate elements presented in Fig. 4 vary in accordance with the harmonic law. The same results were obtained by Fleischmann and Berns (2013).

It should be pointed out that the Lagrange equations were used earlier to study the dynamics of multi-element MTA (Beloev et al., 2015). Application of this method does not allow investigation of the aggregate dynamics, which may change the scheme of aggregation. When considering changes in the aggregate design, it is necessary to create also the movement equations anew. Proposed method of forming the equations of the dynamics of a multi-element MTA (Eqs. 1–5) allows automatic rearrangement of the movement equations according to the aggregate scheme. There was no such a method for studying the dynamics of multi-unit aggregates before; therefore, the results of this study are important and topical.

Conclusion

In terms of compilation of mathematical models of a plane-parallel movement of MTA in a tractor aggregate with an articulated frame, it is necessary to take into account the pseudo-velocity of the first tractor half-frame as a generalised coordinate. In this case, the dynamic model has one degree of freedom, and the mathematical model consists of one dynamics equation and 29 coordinate equations with dependent variations.

While the MTA is moving across the field, there is a deviation in its elements from the rectilinear path approximated by a sinusoid. Obtained model solution made it possible to establish the relationship of the dynamics and MTA energy. Proposed mathematical model may be used to study the dynamics not only of agricultural multi-element aggregates but also trailers used in the automotive and forestry industries.

Aforementioned method for determining the rotation speed of tractor wheels allows exploration of slip dynamics for each wheel individually at the current time (the instantaneous value). The results of the experimental investigations have proved the efficiency of used measuring system for the purposes of field and laboratory tests of tractors and MTAs.

The average value of slipping of the tractor propulsors $\delta = 5\%$; the maximum value of the slipping $\delta_{max} = 11.5\%$ does not exceed the maximum value of agrotechnical requirements. At a fixed bending angle of the tractor half-frames $\psi = 10°$, the aggregate turning radius $r_p = 16.25$ m. As a result of the presented research, the discrepancy between the values of the movement trajectories of the tractor, seed tank and sowing machine did not exceed 4%.

Results and discussion

BELOEV, H. – BORISOV, B. – ADAMCHUK, V. – PETRYCHENKO, I. 2015. Theory of movement of the combined seeding unit. In Agriculture and Agricultural Science Procedia, vol. 7, pp. 21–26.

BLUNDELL, M. – HARTY, D. 2004. The Multibody System Approach to Vehicle Dynamics. Elsevier, 518 pp.

BULGAKOV, V. – ADAMCHUK, V. – GOROBEY, A – OLT, J. 2016. Theory of the oscillations of a toothed disc opener during its movement across irregularities of the soil surface. In Agronomy Research, vol. 14, no. 3, pp. 711–724.

BULGAKOV, V. – KUVACHOV, V. – NOZDROVICKÝ, L. – FINDURA, P. – SMOLINSKYI, S. – IHNATIEV, Y. 2018. The study of movement of the wide span tractor-based field machine unit with power method of its control. In Acta Technologica Agriculturae, vol. 21, no. 4, pp. 160–165.

CHIEH, C. 1995. Dynamic modeling of articulated vehicles for automated highway systems. In Proceedings of the American Control Conference, Seattle, USA, pp. 653–657.

FRANKE, M. – ZAICZEK, T. – ROBENACK, K. 2012. Simulation of nonholonomic mechanical systems using algorithmic differentiation. In IFAC Proceedings Volumes, vol. 45, no. 2, pp. 228–233.

DENHAM, J. – ASMUSSEN, H. – MILLNER, J. – RADNETTER, S. 2007. Networked real-time kinematic corrections for high accuracy machine guidance – dynamic accuracy trials. In International Global Navigation Satellite Systems Society IGNSS Symposium 2007. The University of New South Wales, Sydney, Australia.

FLEISCHMANN, T. – BERNS, K. 2013. Trajectory planning and lateral control for agricultural guidance applications. In 8th International Conference on Information Technology and Applications, pp. 128–133.

IVANOVS, S. – BULGAKOV, V. – NADYKTO, V. – KUVACHOV, V. 2018. Theoretical investigation of turning ability of two-machine sowing aggregate. In Engineering for Rural Development, vol. 17, pp. 314–322.

KARKEE, M. – STEWARD, B. 2010. Study of the open and closed loop characteristics of a tractor and a single axle towed implement system. In Journal of Terramechanics, vol. 47 , pp. 379–393.

LILJEDAHL, J. B. – TURNQUIST, P. K. – SMITH, D. W. – HOKI, M. 1996. Tractors and Their Power Units. 4th ed., New York, USA, 459 pp. ISBN-13:978-1-4684-6634-8.

SIEW, K. – KATUPITIYA, J. – EATON, R. – POTA, H. 2009. Simulation of an articulated tractor-implement-trailer model under the influence of lateral disturbances. In IEEE/ASME International Conference on Advanced Intelligent Mechatronics (AIM 2009), Singapore, pp. 951–956.

WERNER, R. 2012. Dynamic modeling and path tracking control for a farm tractor towing an implement with steerable wheels and steerable drawbar. In Commercial Vehicle Technology Symposium, 13 – 15 March 2012, Kaiserslautern, Germany, pp. 241–250.

YAHYA, A. 2000. Tractor with Built-in DGPS for Mapping Power and Energy Demand of Agricultural Field Operations in Malaysia. In UPM Research Report 1997–2000, vol. 2, 131 pp.

YAHYA, A. – ZOHADIE, M. – KHEIRALLA, A. F. – GEW, S. K. – WEE, E. B. 2004. Precision system for mapping terrain trafficability, tractor-implement performance and tillage quality. In Proceedings of the 7th International Conference on Precision Agriculture and Other Precision Resources Management. 25 – 28 July 2004, Minneapolis, USA, pp. 23–41.