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INFLUENCE OF CHANGE OF HYDRAULIC MACHINE CONTROL PARAMETER DURING BRAKING OF THE TRACTOR WITH THE CONTINUOUSLY VARIABLE TRANSMISSION

Досліджено процес гальмування колісного трактора з безступінчастою гідрооб'ємно-механічною трансмісією. Визначено вплив форми зміни параметру регулювання гідромашини гідрооб'ємної передачі на кінематичні, силові та енергетичні показники колісного трактора при гальмуванні. Доведено доцільність використання вигнутої форми зміни параметру регулювання гідромашини при гальмуванні за рахунок більш ефективної інтенсивності зміни, що призводить до зменшення часу гальмування та гальмівного шляху.

Ключові слова: *колісний трактор, безступінчаста трансмісія, динамічна модель, форма зміни, параметр регулювання гідромашини при гальмуванні.*

1. Introduction

Today in the world of tractor construction there is a tendency in the application of continuously variable transmissions instead of mechanical stepped transmissions. Among continuously variable transmissions, the most demanded are transmissions with variator, hydrostatic mechanical and electromechanical transmissions.

Particularly common for tractors is, of course, continuously variable two-stroke hydrostatic mechanical transmission (GMT). Today, in the world market there are such models of tractors with GMT, as John Deere, Case IH, John Deere, Fendt, Massey Ferguson [1–5]. As for the Ukrainian market, in the mass production there are models made by the joint efforts of the scientists of the National Technical University «Kharkiv Polytechnic Institute» and JSC «Kharkiv Tractor Plant» (Ukraine) – KhTZ-21021, KhTZ-240K with GMT-1C.

An important factor that influences the operation of the tractor with GMT when performing transportation operations is the tractor's control (i. e. change of regulation parameter of the hydraulic machine of hydrostatic mechanical gear (HMG)). After all, it is known that during the braking phase an important factor is the braking time and braking distance, which characterize the safety of the tractor's movement. That is why the study of the influence of the shape of the hydraulic control parameters change during braking is an actual problem.

2. The object of research and its technological audit

The object of research is the braking process of the wheeled tractor KhTZ-21021, which is equipped with continuously variable GMT-1C (Fig. 1).

During the use of the standard braking system, the braking process occurs by pressing the brake and clutch pedals, which kinematically breaks the connection (due to the «clutch») between the hydraulic motor and the sun gear of the differential mechanism. In the case of braking only due to the change of hydraulic machine control parameter, this kinematic relationship is preserved. Special attention should be given to the question of studying the shape of the change in the HMG control parameter (usually a parameter of the hydraulic machine control) versus time that can significantly affect the braking efficiency.

Fig. 1. 3D images of prospective GMT-1C

3. The aim and objectives of research

The aim of research is a theoretical study of the braking process by using the hypothesis concerning the convex-curved shape of the change of hydraulic machine control parameter using an example of simulation of the dynamic model of the wheeled tractor KhTZ-21021 with GMT. To achieve this aim, the following tasks must be accomplished:

1. To describe the dynamic model of the wheeled tractor KhTZ-21021 with the description of the engine,

GMT, and also the interaction of the wheels with the supporting surface.

2. To compare the results in the braking process of the tractor, using a linear form with a convex-curved shape of the change of hydraulic machine control parameter.

4. Research of existing solutions of the problem

Analyzing the scientific literature, several research directions have been established, related to the study of the processes occurring in GMT:

– general tendencies of the development of tractor construction with GMT [1–5];

– increase in the GMT efficiency [6–11];

– suggesting a hypothesis regarding the effective acceleration and deceleration of mobile machines (in particular, tractors) [12–15].

In [6], the authors propose to increase the GMT efficiency by adding an air turbine, as the motive of a hydraulic pump, which will increase the efficiency by 17 %. However, such innovative approach has not yet been approved for GMT as part of a wheeled tractor.

The materials [7] study the optimal control of the efficiency balance between the internal combustion engine and the HMG. However, the authors note that in order to achieve optimal control, it is necessary to separately determine the adjustment characteristics of both the hydraulic pump and the hydraulic motor.

It is determined in [8] that it is advisable to use an uncontrolled hydraulic motor for efficient GMT operation in the hydraulic branch. Such assertion the author will get thanks to the study of the selection of a successful inclination angle of the washer of the hydraulic motor. During the selection process it is found that in some cases, especially at low speeds and torques, the hydraulic system can produce greater efficiency with a smaller inclination angle of the hydraulic motor washer.

The work [9] is devoted to the investigation of the HMG effectiveness in the composition of a land-harvesting machine. In this paper, the author, using mathematical modeling and experiment, establishes the HMG effectiveness and machine control in general using one and two hydraulic motors. Identifying the research, the author notes that at this stage of the study, attention should be paid to the dependence of the loss characteristics on the torque of the load and the regulation parameters of hydraulic machines.

In [11], the author investigates the influence of the HMG special zone on the technical and economic performance of a wheeled tractor with a continuously variable GMT.

In [12, 13], the author, with the help of reducing the generalized energy balance equation, proposes a mathematical model that gives a near-optimal functional dependence of the HMG hydraulic control parameter and the acceleration time of mobile machines. In [14], rational changes in the hydraulic machines control parameters for tractors with GMT, working with «input differential» and «output differential» are established. The materials of this work solve such problem as increasing the technical and economic indicators of the machine and tractor unit with GMT during the plowing operation in the dispersal process.

In the materials [15], the authors give a hypothesis and experimentally confirm it on an imitation stand in the case of use of a convex-curved shape of the change in the parameters of the HMG hydraulic machine control in the process of acceleration and deceleration of mobile machines with a continuously variable GMT. The essence of the hypothesis is that with a convex dependence of the GMT control parameter on time (in comparison with the linear dependence), the most efficient acceleration of the machine is provided, and in the curved form the most effective braking is provided.

Thus, the results of the analysis make it possible to draw a conclusion about the expediency of carrying out a theoretical study of the braking process using the hypothesis of a convex-curved shape of the change of hydraulic machine control parameter by simulating the dynamic model of the wheeled tractor KhTZ-21021 with GMT-1C.

5. Methods of research

In the paper, the Runge-Kutta method is used to solve differential equations describing the properties of the internal combustion engine, GMT, braking system, as well as the interaction of wheels with the supporting surface during the braking process.

The mathematical description of the operation of the engine and GMT-1C, according to the GMT structural scheme (Fig. 2), has the following form:

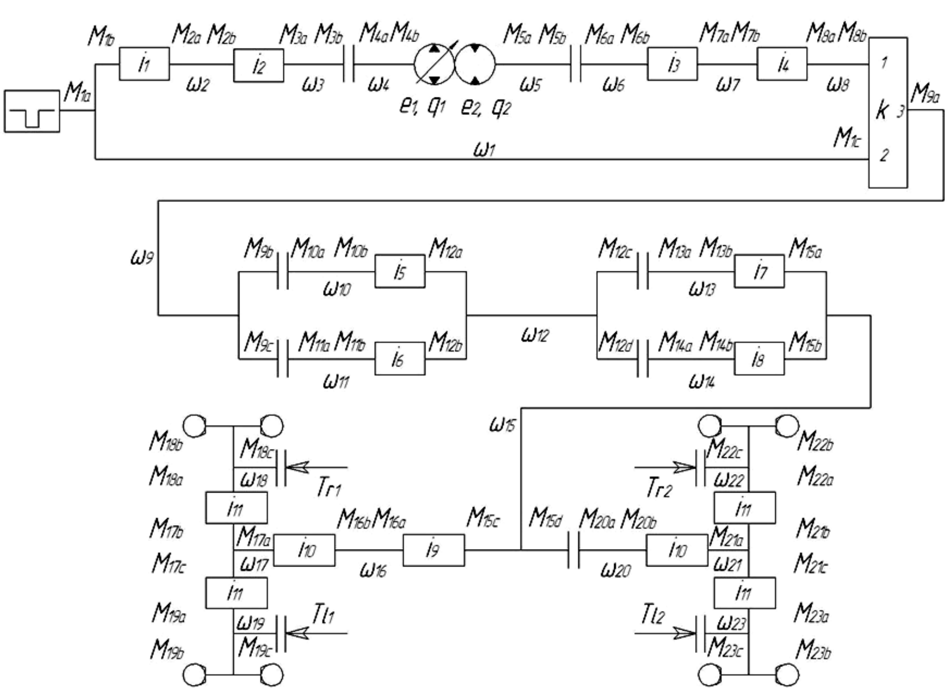

Fig. 2. Structural scheme of GMT-1C

– a system of equations describing the change in the angular acceleration of the transmission elements:

$$\begin{cases}
\frac{d\omega_1}{dt}\cdot i_1-\frac{d\omega_2}{dt}=0;\frac{d\omega_2}{dt}\cdot i_2-\frac{d\omega_3}{dt}=0;\ \frac{d\omega_3}{dt}-\frac{d\omega_4}{dt}=0;\\
e_1\cdot q_1\cdot\frac{d\omega_4}{dt}+q_1\cdot\omega_4\cdot\frac{de_1}{dt}-e_2\cdot q_2\cdot\frac{d\omega_5}{dt}-q_2\cdot\omega_5\cdot\frac{de_2}{dt}=\\
=\left(\frac{K_{1y}}{\mu}\cdot\left(1+C_{1y}\cdot|\omega_4|\right)+\frac{K_{2y}}{\mu}\cdot\left(1+C_{2y}\cdot|\omega_5|\right)\right)\cdot\frac{d(\Delta P)}{dt}+\\
+\left(\frac{K_{1y}}{\mu}\cdot C_{1y}\cdot\frac{d|\omega_4|}{dt}+\frac{K_{2y}}{\mu}\cdot C_{2y}\cdot\frac{d|\omega_5|}{dt}\right)\cdot\Delta P+\\
+\frac{V_{0*}}{2\cdot\pi\cdot E(g^*)}\cdot\left(|\omega_4|+|\omega_5|\right)\cdot\frac{d(\Delta P)}{dt}+\\
+\frac{V_{0*}}{2\cdot\pi\cdot E(g^*)}\cdot\left(\frac{d|\omega_4|}{dt}+\frac{d|\omega_5|}{dt}\right)\cdot\Delta P;\\
\frac{d\omega_5}{dt}-\frac{d\omega_6}{dt}=0;\frac{d\omega_6}{dt}\cdot i_3-\frac{d\omega_7}{dt}=0;\frac{d\omega_7}{dt}\cdot i_4-\frac{d\omega_8}{dt}=0;\\
\frac{d\omega_8}{dt}-k\cdot\frac{d\omega_1}{dt}+(k-1)\cdot\frac{d\omega_9}{dt}=0;\\
\frac{d\omega_{10}}{dt}\cdot i_5-\frac{d\omega_{12}}{dt}=0;\frac{d\omega_{11}}{dt}\cdot i_6-\frac{d\omega_{12}}{dt}=0;\\
\frac{d\omega_{13}}{dt}\cdot i_7-\frac{d\omega_{15}}{dt}=0;\frac{d\omega_{14}}{dt}\cdot i_8-\frac{d\omega_{15}}{dt}=0;\\
\frac{d\omega_9}{dt}-\frac{d\omega_{10}}{dt}=0,\frac{d\omega_{12}}{dt}-\frac{d\omega_{14}}{dt}=0\ (\mathrm{I}-range);\\
\frac{d\omega_9}{dt}-\frac{d\omega_{11}}{dt}=0,\frac{d\omega_{12}}{dt}-\frac{d\omega_{14}}{dt}=0\ (\mathrm{II}-range);\\
\frac{d\omega_9}{dt}-\frac{d\omega_{10}}{dt}=0,\frac{d\omega_{12}}{dt}-\frac{d\omega_{13}}{dt}=0\ (\mathrm{III}-range);\\
\frac{d\omega_9}{dt}-\frac{d\omega_{11}}{dt}=0,\frac{d\omega_{12}}{dt}-\frac{d\omega_{13}}{dt}=0\ (\mathrm{IV}-range);\\
\frac{d\omega_{15}}{dt}\cdot i_9-\frac{d\omega_{16}}{dt}=0;\frac{d\omega_{16}}{dt}\cdot i_{10}-\frac{d\omega_{17}}{dt}=0;\\
\frac{d\omega_{17}}{dt}\cdot i_{11}-\frac{d\omega_{18}}{dt}=0;\frac{d\omega_{17}}{dt}\cdot i_{11}-\frac{d\omega_{19}}{dt}=0;\\
\frac{d\omega_{15}}{dt}-\frac{d\omega_{20}}{dt}=0;\frac{d\omega_{20}}{dt}\cdot i_{10}-\frac{d\omega_{21}}{dt}=0;\\
\frac{d\omega_{21}}{dt}\cdot i_{11}-\frac{d\omega_{22}}{dt}=0;\frac{d\omega_{21}}{dt}\cdot i_{11}-\frac{d\omega_{23}}{dt}=0,
\end{cases}\qquad(1)$$

where $\frac{d\omega_i}{dt}$ – the angular acceleration of the link; i_j – gear ratio; e_1, e_2 – parameters of regulation of HMG hydraulic machines (1 – hydraulic pump; 2 – hydraulic motor); q_1, q_2 – maximum productivity of hydraulic machines; K_{iy}, C_{iy} – loss factors for the hydraulic pump ($i=1$) and for the hydraulic motor ($i=2$); $i=2$ – coefficient of dynamic viscosity; ω_4, ω_5 – angular velocities of the shaft of hydraulic machines and hydraulic motor; ΔP – differential pressure in the HMG; V_{0*} – the volume of the compressed liquid; $E(g^*)$ – elasticity modulus of the working fluid depends on the percentage of gas content g^*; k – internal gear ratio of the planetary series;

– GMT power parameters are described by the system of the following equations:

$$\begin{cases}
J_{dvs}\cdot\frac{dw_{dvs}}{dt}=M_e-M_{0A};M_{1b}\cdot\eta_1^{\Theta\cdot sign(N_{1b})}+i_1\cdot M_{2a}=0;\\
M_{2b}\cdot\eta_2^{\Theta\cdot sign(N_{2b})}+i_2\cdot M_{3a}=0;M_{3b}+M_{4a}=0;\\
M_{4b}-e_1\cdot q_1\cdot\Delta P=-\Delta M_1\cdot sign(\omega_4);\\
M_{5a}+e_2\cdot q_2\cdot\Delta P=-\Delta M_2\cdot sign(\omega_5);\\
M_{6b}\cdot\eta_3^{\Theta\cdot sign(N_{6b})}+i_3\cdot M_{7a}=0;\\
M_{7b}\cdot\eta_4^{\Theta\cdot sign(N_{7b})}+i_4\cdot M_{8a}=0;M_{5b}+M_{6a}=0;\\
M_{8b}\cdot\eta_{13}^{\Theta\cdot sign(N_{8b})}+M_{1c}\cdot\eta_{23}^{\Theta\cdot sign(N_{1c})}+M_{9a}=0;\\
M_{8b}\cdot k\cdot\eta_{13}^{\Theta\cdot sign(N_{8b})}+M_{1c}\cdot\eta_{23}^{\Theta\cdot sign(N_{1c})}=0;\\
M_{10b}\cdot\eta_5^{\Theta\cdot sign(N_{10b})}+i_5\cdot M_{12a}=0;\\
M_{11b}\cdot\eta_6^{\Theta\cdot sign(N_{11b})}+i_6\cdot M_{12b}=0;\\
M_{13b}\cdot\eta_7^{\Theta\cdot sign(N_{13b})}+i_7\cdot M_{15a}=0;\\
M_{14b}\cdot\eta_8^{\Theta\cdot sign(N_{14b})}+i_8\cdot M_{15b}=0;\\
M_{9b}+M_{10a}=0,M_{12d}+M_{14a}=0,\\
M_{9c}=M_{11a}=M_{12c}=M_{13a}\ (\mathrm{I}-range);\\
M_{9c}+M_{11a}=0,M_{12d}+M_{14a}=0,\\
M_{9b}=M_{10a}=M_{12c}=M_{13a}\ (\mathrm{II}-range);\\
M_{9b}+M_{10a}=0;M_{12c}+M_{13a}=0;\\
M_{9c}=M_{11a}=M_{12d}=M_{14a}\ (\mathrm{III}-range);\\
M_{9c}+M_{11a}=0,\ M_{12c}+M_{13a}=0,\\
M_{9b}=M_{10a}=M_{12d}=M_{14a}\ (\mathrm{IV}-range);\\
M_{15c}\cdot\eta_9^{\Theta\cdot sign(N_{15c})}+i_9\cdot M_{16a}=0;\\
M_{17b}\cdot\eta_{11}^{\Theta\cdot sign(N_{17b})}+i_{11}\cdot M_{18a}=0;\\
M_{17c}\cdot\eta_{11}^{\Theta\cdot sign(N_{17c})}+i_{11}\cdot M_{19a}=0;\\
(M_{15d}+M_{20a})\cdot Y=0;M_{15d}=0,Y=0;\\
M_{20a}=0,Y=0;M_{20b}\cdot\eta_{10}^{\Theta\cdot sign(N_{20b})}+i_{10}\cdot M_{21a}=0;\\
M_{21b}\cdot\eta_{11}^{\Theta\cdot sign(N_{21b})}+i_{10}\cdot M_{22a}=0;\\
M_{21c}\cdot\eta_{11}^{\Theta\cdot sign(N_{21c})}+i_{11}\cdot M_{23a}=0;M_{1a}+M_{1b}+M_{1c}=0;\\
M_{2a}+M_{2b}=0;M_{3a}+M_{3b}=0;\\
M_{4a}+M_{4b}=0;M_{5a}+M_{5b}=0;\\
M_{6a}+M_{6b}=0;\ M_{7a}+M_{7b}=0;\\
M_{8a}+M_{8b}=0;\ M_{9a}+M_{9b}+M_{9c}=0;\\
M_{10a}+M_{10b}=0;M_{11a}+M_{11b}=0;\\
M_{12a}+M_{12b}+M_{12c}+M_{12d}=0;\\
M_{13a}+M_{13b}=0;M_{14a}+M_{14b}=0;\\
M_{15a}+M_{15b}+M_{15c}+M_{15d}=0;\\
M_{16a}+M_{16b}=0;M_{17a}+M_{17b}+M_{17c}=0;\\
M_{18a}+M_{18b}+M_{18c}\cdot T_{r1}=0;\\
M_{19a}+M_{19b}+M_{19c}\cdot T_{l1}=0;M_{20a}+M_{20b}=0;\\
M_{21a}+M_{21b}+M_{21c}=0;\\
M_{22a}+M_{22b}+M_{22c}\cdot T_{r2}=0;M_{23a}+M_{23b}+M_{23c}\cdot T_{l2}=0;\\
T_{r1}=T_{r2}=T_{l1}=T_{l2}=1;\\
J_{Tr1}\cdot\frac{d\omega_{18}}{dt}=M_{Tr1}-M_{18c};J_{Tl1}\cdot\frac{d\omega_{23}}{dt}=M_{Tl1}-M_{23c};\\
J_{Tr2}\cdot\frac{d\omega_{22}}{dt}=M_{Tr2}-M_{22c};J_{Tl2}\cdot\frac{d\omega_{23}}{dt}=M_{Tl2}-M_{23c},
\end{cases}\qquad(2)$$

where J_{dvs} – the moment of inertia of the flywheel masses of the internal combustion engine brought to the shaft; $\frac{dw_{dvs}}{dt}$ – angular acceleration of the crankshaft of the internal combustion engine; M_e – effective torque of the internal combustion engine; M_{0A} – moment of resistance to movement; M_{nm} – moments on the GMT links; m – the index number coincides with the number of the angular velocity of the link; n – indexes-letters correspond to moments at the ends of links [16]; η_j – reducer efficiency; Θ – loss factor in gearing ($\Theta=0$ – without losses, $\Theta=-1$ – taking into account losses in gearing); N_{nm} – the power transmitted by the GMT links (the product of the angular velocities at the appropriate moments, taking into account the sign, give the magnitude and direction of the power flows on specific links and elements of the GMT) [16]; η_{13}, η_{23} – efficiency in gearing of the sun-satellite and epicycle-satellite with the stopped carrier, determining the loss of moments; ΔM_1, ΔM_2 – moment losses in hydraulic machines, calculated, for example, according to the Gorodetsky mathematical loss model [16], as a function of the control parameters, angular velocity of hydraulic machine shafts, working volumes q_1, q_2 and pressure drop ΔP; J_i – inertia moment reduced to the links of the GMT elements; T_{ij} – brake activation parameter ($T_{ij}=1$ – the brake is on, $T_{ij}=0$ – off, $i=r$ – the starboard side, $i=l$ – the port side, $j=1$ – the front axle, $j=2$ – the rear axle); J_{Tij} – inertia moment of the brake link, to which, on the one hand, the driving torque from the transmission is applied, and on the other hand the braking torque M_{Tij} from the braking element; M_{Tij} – the moment created by the braking element (braking torque) [17].

It is known from [16] that the moment of losses in hydraulic machines:

$$\Delta M_i = q_i \cdot \left[\begin{aligned} & K_1 \cdot |\omega_i| \cdot (1 + K_2 \cdot \overline{e}_i^2) + \\ & + \frac{K_5 \cdot (1 + K_4 \cdot |\overline{e}_i|)}{(1 + K_3 \cdot |\omega_i| \cdot D_{qi})} \cdot \Delta P + \\ & + \frac{K_8 \cdot (1 + K_7 \cdot |\overline{e}_i|)}{(1 + K_6 \cdot |\omega_i| \cdot D_{qi})} \end{aligned} \right], \qquad (3)$$

where $K_1, K_2, \ldots K_8$ – the coefficients of hydromechanical losses [16]; D_{qi} – characteristic size of the hydraulic machine,

$$D_{qi} = \sqrt[3]{2 \cdot \pi \cdot q_i}.$$

The braking torque is calculated as follows:

$$M_{Tij} = M_{T0ij}(P_{ij}) \cdot sign(-\omega_{Tij}), \qquad (4)$$

where $M_{T0ij}(P_{ij})$ – the absolute magnitude of the braking torque; P_{ij} – working medium pressure; ω_{Tij} – angular speed of rotation of the brake link.

In describing the interaction of wheels with a support surface, the following are used: the d'Alembert's principle, Lagrange equations of the second kind, the partial acceleration method, and the like. In the work, more emphasis is placed on the study of processes in a continuously variable GMT. Therefore, it is quite appropriate to apply the description of the interaction of wheels with the support surface using the equation that reproduces the motion dynamics of a single wheel during the braking process [17]:

$$J_{\Sigma ij} \cdot \frac{d\omega_{ij}}{dt} = M_{xij} - M_{fij} - M_{ij}, \qquad (5)$$

where $J_{\Sigma ij}$ – inertia moment of the rotating masses associated with the wheel; $\frac{d\omega_{ij}}{dt}$ – angular retardation of the wheel; M_{xij} – moment created by the reaction in the longitudinal plane of the wheel; M_{fij} – moment of rolling resistance of the wheel; M_{ij} – moment on the wheel (it should be noted that from the work [17] the braking moment M_{Tij} is component of M_{ij}).

From work [17], the moment M_{xij} created by the reaction in the longitudinal plane of the wheel, which determines the braking efficiency, is calculated using the following equation:

$$M_{xij} = R_{zij} \cdot \varphi_{xij} \cdot r_{dij}, \qquad (6)$$

where R_{zij} – the normal reaction of the road in contact of the tractor wheels with the supporting surface; φ_{xij} – traction coefficient of the tractor wheels with a supporting surface in the longitudinal direction; r_{dij} – radius of the wheels.

Expression (5) with the movement of the wheels of the tractor KhTZ-21021, taking into account the features of the mathematical model of the transmission GMT-1C in the process of braking will be the following form [17]:

$$J_{\Sigma r1} \cdot \frac{d\omega_{18}}{dt} \cdot sign(\omega_{18}) = M_{xr1} - M_{fr1} - M_{18b} \cdot sign(\omega_{18});$$

$$J_{\Sigma l1} \cdot \frac{d\omega_{19}}{dt} \cdot sign(\omega_{19}) = M_{xl1} - M_{fl1} - M_{19b} \cdot sign(\omega_{12});$$

$$J_{\Sigma r2} \cdot \frac{d\omega_{22}}{dt} \cdot sign(\omega_{22}) = M_{xr2} - M_{fr2} - M_{22b} \cdot sign(\omega_{22});$$

$$J_{\Sigma l2} \cdot \frac{d\omega_{23}}{dt} \cdot sign(\omega_{15}) = M_{xl2} - M_{fl2} - M_{23b} \cdot sign(\omega_{23}). \qquad (7)$$

The author [17] notes that in the case of blocking the wheel when the tractor brakes during forward motion, the change in equation (7) to

$$\frac{d\omega_{ij}}{dt} = 0.$$

In addition, with a decrease of M_{Tij} (during the period when the wheel is locked), it is necessary to choose from the condition:

$$J_{\Sigma ij} \cdot \frac{d\omega_{ij}}{dt} =$$

$$= \max(M_{xij} - M_{fij} - M_{ij}, 0) \begin{cases} \omega_{ij} = 0, \text{ if } \omega_{ij} < 0; \\ \frac{d\omega_{ij}}{dt} = 0, \text{ if } \omega_{ij} = 0 \text{ and } \frac{d\omega_{ij}}{dt} < 0. \end{cases} \qquad (8)$$

In [17], the author notes that the braking and further blocking of the wheel is due to a constant change in the vertical reaction of the road R_{zij} and the adherence coefficient φ_{xij}, which in turn is due to the redistribution of the tractor's weight between the axles.

Considering the flat mathematical model of the tractor in the braking process, the author [17] notes that the total normal reaction of the road R_{zij}, taking into account the redistribution of masses during braking of the tractor is determined by the expression):

– on the front wheels R_{zi1} $(R_{zr1} = R_{zl1})$:

$$R_{zi1} = \frac{G \cdot b - F_b \cdot h_b + F_j \cdot h - \sum_{i,j} R_{ZCTij} \cdot f_{ij} \cdot r_{dij}}{2(a+b)}; \quad (9)$$

– on the rear wheels R_{zi2} $(R_{zr2} = R_{zl2})$:

$$R_{zi2} = \frac{G \cdot a + F_b \cdot h_b - F_j \cdot h + \sum_{i,j} R_{ZCTij} \cdot f_{ij} \cdot r_{dij}}{2 \cdot (a+b)}, \quad (10)$$

where G – weight of the tractor; a, b, h – coordinates of the tractor mass center; F_b – force of air resistance; h_b – the distance from the reference surface to the air resistance force; F_j – resistance to acceleration of the tractor; R_{ZCTij} – static load on the tractor wheels; f_{ij} – coefficient of rolling resistance of the wheel.

It is known from [16] that the force of air resistance is determined by the equation:

$$F_b = k_b \cdot F \cdot V^2, \quad (11)$$

where k_b – the coefficient of air resistance; F – drag area; V – tractor speed.

Inertia force acting on the wheeled tractor in the process of braking author [17] determines from the expression:

$$F_j = sign(-\omega_{ij}) \cdot \frac{G \cdot \ddot{X} \cdot \delta_t}{g}, \quad (12)$$

where $\ddot{X}$ – retardation of the tractor along the coordinate axis X; δ_t – factor of the account of masses of the engine and transmission, the running system which are rotating; g – gravitational acceleration.

Fig. 3 shows the design of the forces acting on the wheeled tractor during the braking process. Since the movement of the wheeled tractor occurs in a relatively horizontal section, the inclination angle of the road surface is not taken into account.

Determination of the slowing down of the wheeled tractor along the coordinate axis X, from [16] is determined as follows:

$$\ddot{X} = \frac{\left[-\left(\sum_{i,j} R_{xij} \right) - F_b - \sum_{i,j} R_{zij} \cdot f_{ij} \right] \cdot g}{G \cdot \delta_t}, \quad (13)$$

where R_{xij} – the reaction in the longitudinal plane of the tractor wheel determines the braking efficiency.

Analyzing the scientific paper [17], it is customary to use a coefficient φ_{xij} for estimating the coupling capabilities of the wheel in the longitudinal direction:

$$\varphi_{xij} = R_{xij} / R_{zij}. \quad (15)$$

The braking distance is determined from the following expression:

$$S(t) = \int_0^{\infty} V(t)\,dt = r_k \cdot \int_0^{\infty} \omega_{ij}(t)\,dt. \quad (16)$$

According to GOST 12.2.019-86, the braking distance is calculated by the empirical formula:

$$S \le 0{,}15 \cdot V + \frac{V^2}{116}. \quad (17)$$

Thus, on the basis of equations (1)–(16), a dynamic model of the wheeled tractor KhTZ-21021 with a continuously variable GMT-1C is compiled. This model is implemented in the MATLAB system of the Simulink dynamic modeling subsystem

6. Research results

The results of modeling the braking process of the wheeled tractor KhTZ-21021 with a continuously variable GMT-1C is shown in Fig. 4.

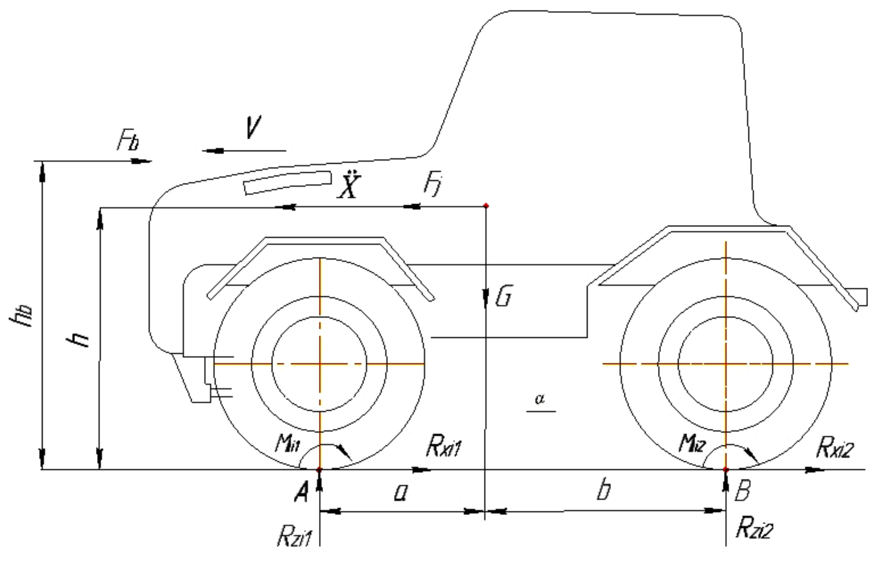

Fig. 3. Scheme of forces acting during the braking of a wheeled tractor

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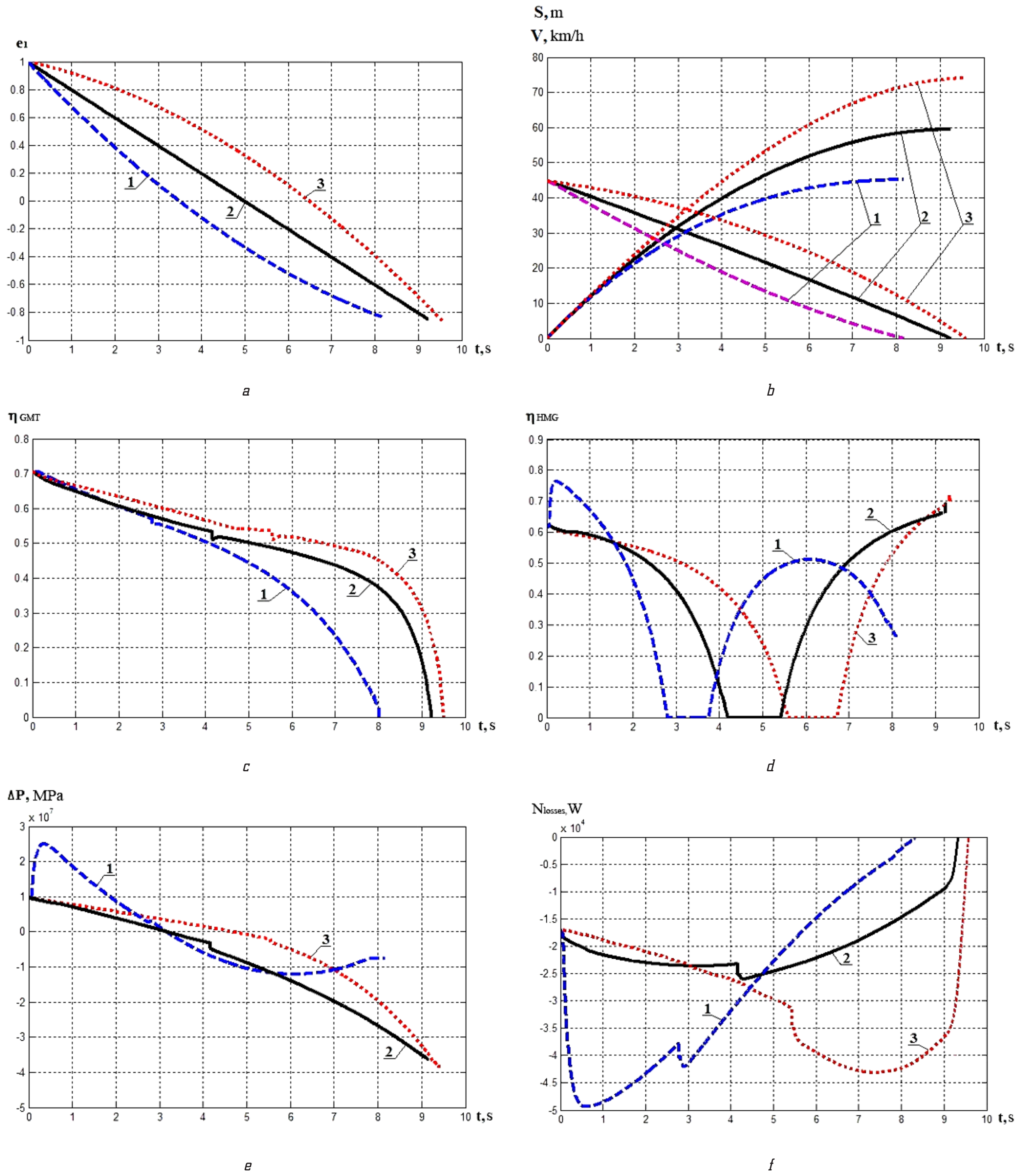

Fig. 4. The results of the theoretical study of the braking process of the tractor KhTZ-21021 with GMT-1C: *a* – the parameter of the hydraulic pump (e_1); *b* – tractor speed (V) and braking distance (S); *c* – GMT efficiency (η_{GMT}); *d* – HMG efficiency of (η_{HMG}); *e* – pressure drop in HMG (ΔP); *f* – power loss (N_{loss}); 1 – convex shape of change; 2 – linear shape of change; 3 – curved shape of change

Thus, the kinematic, power and energy parameters are changed during braking, using the curved, linear and convex forms of the change in the parameter of the HMG hydraulic machine regulation.

7. SWOT analysis of research results

Strengths. The strengths of this research is confirmation of the hypothesis of using curved shape of changing control parameters in GMT hydraulic braking of mobile machines with continuously variable GMT already in a theoretical study of braking a wheeled tractor with GMT.

Making the characteristic evaluation results obtained when implementing a curved, linear and convex shapes of the hydraulic control parameter changes in the HMG should be noted that when compared linear with convex and linear with curved, there are:

– reduction (for linear with curved) of braking time by 11.4 % and an increase (linear with convex) 3.8 %;

– decrease (for linear with curved) of braking distance by 23.3 % and an increase (linear with convex) by 21.7 %.

Thus, it is established that using the curved shape of the change in the hydraulic machine control parameter, the braking time and braking distances decrease. This observation characterizes the increase in the safety of wheeled tractor movement in the performance of transportation operations, which, of course, is the strength of the research.

Identifying of results it is noted that the figures on the braking performance of the tractor, but the form of the hydraulic control parameters change, also significantly affect account the moments of inertia and intensity changes of the hydraulic control parameters.

Weaknesses. The weaknesses of this research are related to the fact that for an objective evaluation of the parameters during braking it is necessary to carry out experimental studies. However, the hypothesis on the form of the change in the hydraulic machine control parameter on an imitation stand is given in [15] and confirmed by experimental tests.

Opportunities. Additional opportunities to achieve the aim of research lie in the generalization of the engine and GMT operation and determination for the particular mobile machine of its optimal forms of variation of the regulation parameter (convex or curved), both for acceleration and for braking. Such more integrated task will make it possible to more clearly formulate the general drive of the wheeled tractor, both during the acceleration process and at the stage of braking of the wheeled tractor with continuously variable GMT.

The introduction of this concept will allow raising the technical level of KhTZ tractors to the world level that will bring additional investment in the development of the state economy.

Threats. Difficulties in implementing the research results are associated with some difficulties in JSC «Kharkiv Tractor Plant». Therefore, during acceleration and braking on wheeled tractors KhTZ-21021 and KhTZ-240K with continuously variable GMT-1C, the implementation of the convex-curved shape of the modification is excluded.

The implementation of research results will incur additional costs to the enterprise, which will be measured, approximately, in 5 % of the value of the tractor.

8. Conclusions

1. The dynamic model of the wheeled tractor KhTZ-21021 is given with a mathematical description of the engine, GMT, as well as the interaction of the wheels with a supporting surface, which differs from the existing ones when calculating the GMT:

– volume of compressed liquid;

– elasticity modulus of the working fluid depends on the percentage of gas content.

2. Comparing the qualitative results (calculation of the area under the graph in Fig. 3, *c, d, f* during the tractor braking, using the linear form of the change with a convex-curved shape of the change in the control parameter of hydraulic machines), let's observe:

– curved with linear: a reduction in the HMG efficiency by 9.3 %, a decrease in the GMT efficiency by 8.7 %, a decrease in the power consumption by 43 %;

– convex with linear: an increase in the HMG efficiency by 11.1 %, an increase in GMT efficiency by 7.4 %, an increase in the power consumption by 50.6 %.

These observations indicate that using the curved shape of the hydraulic control parameter for the tractor KhTZ-21021 during braking, the power losses in the HMG hydraulic link are increasing that is directly related to the efficiency of the braking of the tractor.

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ВЛИЯНИЕ ИЗМЕНЕНИЯ ПАРАМЕТРОВ РЕГУЛИРОВАНИЯ ГИДРОМАШИНЫ В ПРОЦЕССЕ ТОРМОЖЕНИЯ ТРАКТОРА С БЕССТУПЕНЧАТОЙ ТРАНСМИССИЕЙ

Исследован процесс торможения колесного трактора с бесступенчатой гидрообъемно-механической трансмиссией. Определено влияние формы изменения параметра регулирования гидромашины гидрообъемной передачи на кинематические, силовые и энергетические показатели колесного трактора при торможении. Доказана целесообразность использования изогнутой формы изменения параметра регулирования гидромашины при торможении за счет более эффективной интенсивности изменения, что приводит к уменьшению времени торможения и тормозного пути.

Ключевые слова: колесный трактор, бесступенчатая трансмиссия, динамическая модель, форма изменения, параметр регулирования гидромашины при торможении.

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