INVESTIGATING THE INFLUENCE FORM AND INTENSITY OF THE ACCELERATING CHARACTERISTIC MACHINE WITH A DUAL-FLOW NON-REVERSIBLE TRANSMISSION

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Abstract - Now, when tractors are being made, there are three ways to design the way the transmission is installed on agricultural tractors. This is based on looking at how well machine-tractor units with mechanical and continuously variable dual-flow transmissions work. When looking into ways to improve how machine-tractor units with HMCVT work, it should be noted that this process is directly related to improving design parameters or forming rational forms of changing the control parameters of hydraulic machines. This article's aim is to improve how well a machine-tractor unit works. It does this by looking at how the shape (for example, different types of convexity) and how strong the acceleration is for a two-stream non-reversible HMCVT when a wheeled tractor is used for farming in arable areas. The goal was achieved by using a mathematical model of a machine-tractor unit in Matlab. This model took into account the traction operation mode, namely plowing. The goal was achieved by using a mathematical model of a machine-tractor unit in Matlab. This model took into account the traction operation mode, namely ploughing. The work presented here is important because it helps to understand how changes in the control parameter of a hydraulic machine (part of a dual-flow non-reversible HMCVT) affect the performance of a wheeled tractor. The work is new because it is the first time that the relationship between the shape and intensity of the change in the control parameter of the hydraulic machine has been established when studying the acceleration characteristic of the HMCVT of a tractor.

Keywords: Wheeled tractor, Transmission, Acceleration, Operational indicators, Efficiency, Hydraulic pump adjustment, Intensity, Matlab modeling.

1. Introduction

By assessing the factors that cause environmental degradation through the emission of polluting gases, work is underway to introduce alternative energy sources for both specialised and transport vehicles. One of the ways in which the industry is developing is through the introduction of dual-flow transmission units, which is particularly noticeable in agricultural machinery.

Meeting speed and energy requirements while complying with operational standards is the primary objective of any modernisation of agricultural equipment. Therefore, from the initial stage of designing the power plant (combustion engine and transmission), the conditions are created to increase the productivity and efficiency of agricultural equipment in a sustainable mode of operation. However, transient processes, i.e. acceleration and deceleration of vehicles, cannot be neglected, especially when studying a power plant with a dual-flow transmission.

The creation and effective use of modern tractors with ploughs is the basis for the development of the agro-industrial complex of any state. Today's farms use wheeled tractors equipped with conventional continuously variable transmissions (CVTs), in particular hydrovolumetric mechanical transmissions (HMCVT). The use of tractors with CVT is explained by undoubted advantages compared to transmissions: smooth running, improved ergonomics during technological work, automation of control, etc.

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2. State of Transmission Design Research

It is known from [1] that agricultural tractors have different transmissions, which are divided into stepped (mechanical operating principle) and stepless (hydraulic and/or electrical operating principle) transmissions according to the principle of torque conversion [2, 3]. The introduction of electro-mechanical (or hybrid) transmissions on agricultural tractors is becoming increasingly popular due to a significant leap in electric drive technology. Like the automotive industry [4], the tractor industry uses hybrid technologies based on sequential or parallel operation. For example, in [5], structural solutions for a full hybrid transmission are investigated for use in agricultural tractors. The main operating modes considered are hybrid mode, fully electric mode, combustion engine only mode and parallel hybrid mode. In [6] it is proposed to use a serial sequential hybrid transmission instead of the CVT of a tractor, but taking into account the recuperation technologies that occur when the vehicle is used in agricultural areas. It is also worth mentioning the works [7, 8], which present conceptual diagrams of the designs of electromechanical transmissions for agricultural tractors, but the main drawback of the proposed technical solutions is that their implementation is carried out on tractors of low or medium power (up to 130 HP).

Stepped mechanical transmissions were the most popular transmissions used on wheeled tractors in the 20th century. This trend was explained by the fact [9] that they were the most structurally proven, relatively simple and reliable to operate, relatively high efficiency and lower cost. Their main disadvantages are ergonomic problems and inefficient use of the combustion engine's power. Today, mechanical transmissions are represented by PowerShift models (shifting under full or partial load (Semi-PowerShift)) and dual-clutch transmissions [10]. There are a number of publications [11, 12] devoted to the study of the operation of double-clutch transmissions on agricultural tractors. However, the use of this type of transmission on high power tractors is limited by the inability to adapt to the variable load that occurs during traction work.

The introduction of HMCVT on high horsepower wheeled tractors (over 130 hp) began back in the 20th century. As a result, the technology is now well established and widespread, with this type of transmission being used as standard on almost all models from the world's tractor manufacturers.

The basic method of studying the speed control characteristics of HMCVT is theoretical analysis based on the methods of evaluating stepped mechanical transmissions [13]. For example, in [13], based on the theoretical analysis of the speed range control characteristics, the relationship between the degree of linearisation of the speed control characteristics and the range of change of the HMCVT output speed was established. In [14], the relationship between the transmission ratio and the volume coefficient was established using a theoretical model of the output speed. It is also worth mentioning [15], which proposed a method to study the variation of the speed control characteristics based on the design characteristics of the HMCVT operation. In [16], a design model is proposed for the optimisation of HMCVT parameters using a genetic algorithm to implement the optimisation parameters of the transformation. In [17], a method combining speed control characteristic analysis, regression-based correction and an improved particle swarm optimization algorithm (I-PSO) is proposed. The method outlined improves the speed control characteristics of continuously variable transmissions for agricultural machinery to meet the technical and engineering requirements of precision farming and intelligent agriculture.

In order to ensure reliable and efficient operation of agricultural machinery with CVT, it is necessary to correctly coordinate the control characteristics of the "engine-transmission" system [18]. It should be noted that in order to increase the mobility of agricultural machinery for different purposes, which is characterised by its mobility indicators when operating in different areas (traction and transport work), tractors with transmissions are able to transmit a large part of the power through the hydraulic branch, i.e. hydro volumetric transmission. This state of affairs leads to the complexity of mathematical modeling and development of a control system for such a transmission [19]. Linearization of the control systems of the HMCVT leads to the loss of information, which ensures inaccuracy of regulation and a decrease in the technical and operational indicators of the tractor. The method of controlling the displacement of an axial piston hydraulic pump varies from manufacturer to manufacturer. There are three ways to modify the control parameters of hydraulic machines: by mechanical modification, hydraulic modification and electrical modification (using the ISO 11783 regulations). As stated in [20], mechanical operation with the control disc of the hydraulic machine is not found in hydraulic pumps with a working volume of no more than 45-50 cm^3. To achieve automation of the control process in this case, a hydraulic control method is used, i.e. the control system can intervene during the operation of the hydraulic machine.

In turn, the hydraulic motor is controlled by the angle of inclination of the adjusting disc a, the principle of its operation is similar to the operation of a hydraulic pump. As stated in [21], the HMCVT control system should provide increased productivity, accuracy and quality of work, comfort and safety for the driver, as well as reduce the tractor's energy consumption and environmental pollution. In [22], a control architecture using proportional-integral transmission control is presented, which simultaneously ensures smooth gear ratio changes and high efficiency of the entire transmission system. In [23], a variable control algorithm is proposed for an agricultural tractor with an "output differential" HMCVT. The auxiliary gear shift control algorithm, including hysteresis compensation, clutch disengagement time delay and the duration of HMCVT during acceleration, was evaluated by conducting bench-top experimental studies. The experimental results showed that the auxiliary gear shift was performed smoothly without torque interruption, even in the acceleration state when there was a large change in inertial torque. In [24], an active disturbance suppression control algorithm with feedforward compensation for the speed ratio is presented. The active disturbance suppression control with feedforward compensation is effective in speed ratio control. It reduces the speed ratio error of the continuously variable HMCVT and has good adaptability. An integrated control strategy for the "engine-transmission" system is proposed in [25], taking into account the fluctuations in transmission efficiency and the problem of the response delay of the continuously variable transmission combined with the complex operating modes of the tractor. Based on the estimation of the working resistance, a fuzzy algorithm is used to develop the throttle compensation law. Considering the maximum driving power of the tractor as the variable speed control target, the efficiency model of the HMCVT is established and the control law of the gear ratio of the HMCVT with the maximum driving power of the tractor under any working conditions is determined. Based on the wavelet neural network of the proportional-integral-derivative algorithm, the control law of the HMCVT speed control is developed and the proportional-integral-derivative control parameters are adjusted in real time during the control process. The strategy outlined improves the stability of the tractor speed and ensures the quality of work, thereby improving the tractor's ability to adapt to complex working conditions. In [26], an integrated control algorithm for the combustion engine and the HMCVT of a tractor is proposed with emphasis on improving the operating characteristics of the work. In order to evaluate the control algorithm, simulation modelling is carried out based on traction work. According to the simulation results, it is found that the engine speed is maintained at the desired speed by the engine and HMCVT ratio control.

3. Materials and Methods

3.1. Influence of the Form of Intensity Variation on the Acceleration Characteristics of a Continuously Variable Dual-Flow Transmission

The intensity of change in the acceleration characteristic of the HMCVT is understood as a change in the form of the hydraulic pump control parameter, which affects the change in the technical and operational indicators. The assessment of the intensity of change in the acceleration characteristic is based on a qualitative assessment of the time and path of acceleration of the machine-tractor unit. In addition, the study will use a quantitative assessment, which includes a change in technical and operational indicators by an integral indicator.

The basic form of the change in the hydraulic pump control parameter $e_1(t)$ is linear (Fig. 1). Since the change in the control parameter occurs over a certain time, divide the total acceleration time into 4 normalised components, i.e. ¼ t_{accel}, $\frac{2}{4}$ t_{accel}, ¾ t_{accel}. The next step is to draw a perpendicular from the linear form of the change at the points of the normalised time components obtained. The increase in the value of the perpendicular characterizes the intensity of the acceleration curve. The intensity of the change in the acceleration curve (τ) is calculated as follows:

$$\tau = \int_{t=0}^{t_{accel}} e_1^*(t)\,dt \Big/ \int_{t=0}^{t_{accel}} e_1(t)\,dt\,, \qquad (1)$$

where $e_1(t)$ is linear form of change of the hydraulic pump control parameter; e_1^* (t) is studied forms of change of the hydraulic pump control parameter (items 2 - 6 in Fig. 1) [27].

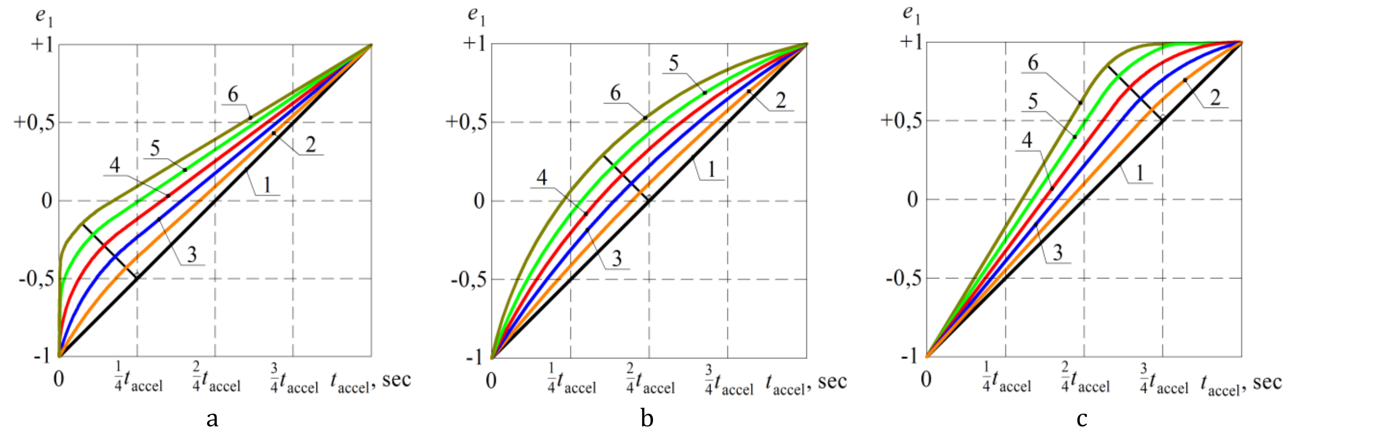


Figure 1: The studied forms of acceleration curve intensity obtained by changing the hydraulic pump control parameter: a - convexity of form at ¼ t_{accel}; b - convexity of form at $2/4$ t_{accel}; c - convexity of form at ¾ t_{accel}; 1 - basic linear shape; 2 - 6 - convex shapes studied

In order to solve these problems, it is necessary to draw a structural diagram of the transmission. Figure 1 shows the structural diagram of the dual-flow HMCVT. Thus, Fig. 1 shows the studied convex forms of the change in the control parameter, which are respectively: pos. 2 - τ = 7.82%; pos. 3 - τ = 15.7%; pos. 4 - τ = 23.7%; pos. 5 - τ = 31.6%; pos. 6 - τ = 40%.

3.2. Wheeled Tractor Simulation Model

The next step is to numerically simulate the movement of a wheeled tractor during ploughing, taking into account different forms of acceleration characteristics [28].

The Dormand-Prince numerical method (ode5) with a step of 0.001 is used to simulate the movement of a machine-tractor unit (Fig. 2). To implement the ploughing operation with a wheeled tractor, work with a five-body mounted plough with a working depth of 0.3 m is simulated. Note that the ploughing characteristic is linked to the increase in the speed of movement V, which in turn depends on the shape and intensity of the variation in the hydraulic pump control parameter $e_1(t)$. In this study, the combustion engine is operated with full fuel supply. It should also be noted that the tyre pressure of the wheeled tractor is 80 kPa, which is set by the manufacturer when the tyres are used during ploughing.

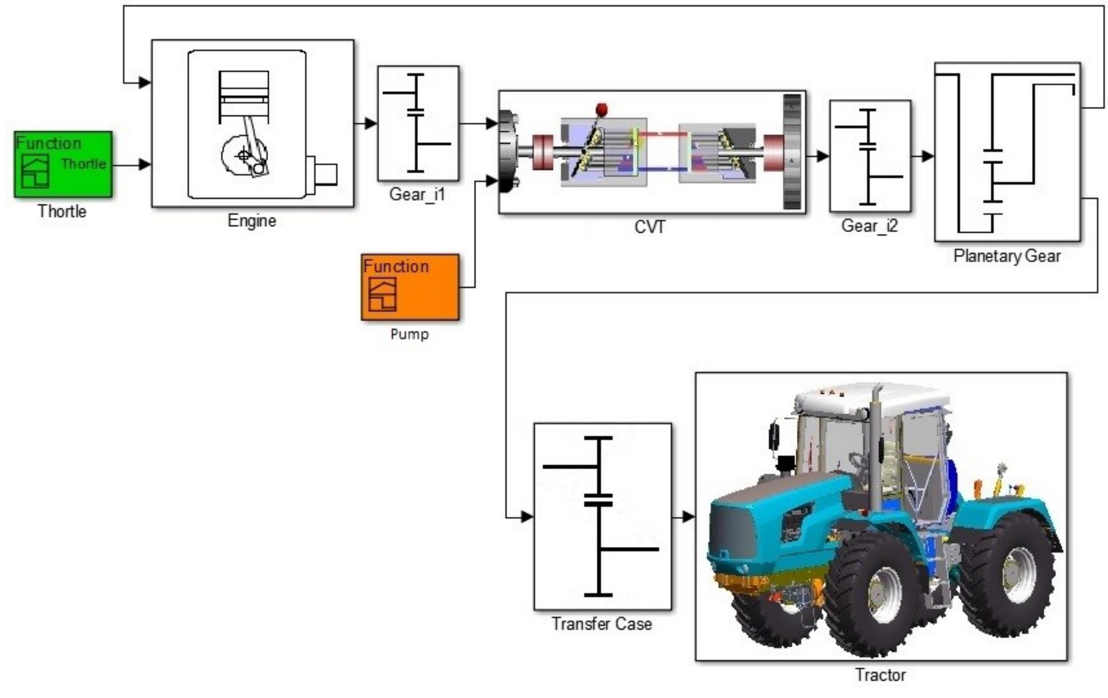


Figure 2: Implementation of a machine-tractor unit model in MATLAB/SimuLink

Since the studied system "Engine - HMCVT" is complex and involves the calculation of several differential equations, a transient (oscillating) process is formed when using any numerical method. Therefore, in order to simplify the obtained values of technical and operational indicators, we average them using the Butterworth filter.

3.3. Evaluation of the Form of Change in the Hydraulic Machine Control Parameter as Part of the Wheeled Tractor Transmission

Solving the problem of the fastest acceleration and braking is important because its implementation makes it possible to increase the technical, economic and ergonomic indicators, as well as having a significant impact on the safety of wheeled tractors [29].

General relationships that relate the variation of the control parameter $e_j(V)$in the j-th range to the speed V of the mobile machine with the HMCVT and vice versa – V from $e_j(V)$:

$$e_j\left(V\right)\cdot sign\frac{d}{dV}\left[e_j(V)\right]=2\frac{V-V_{j\min}}{V_{j\max}-V_{j\min}}-1\,; \tag{2}$$

$$V=\frac{\left[\begin{array}{c}e_j(V)\left(V_{j\max}-V_{j\min}\right)\cdot\mathrm{sign}\frac{d}{dV}\left[e_j(V)\right]+\\ +V_{j\max}+V_{j\min}\end{array}\right]}{2}. \tag{3}$$

For the HMCVT with a non-reversible control characteristic (e=±1), if at the start of the tractor movement at t = 0, e(0) = -1 and at each j-th speed range, all the $V_j(0) = 0$ dependencies (2 - 3) are of the form:

$$e_j\left(V\right)=\left(2\frac{V}{V_{j\max}}-1\right)\cdot\mathrm{sign}\frac{d}{dV}\left[e_j(V)\right]; \tag{4}$$

$$V=\frac{V_{j\max}}{2}\left[e_j(V)\cdot\mathrm{sign}\frac{d}{dV}\left[e_j(V)\right]+1\right]. \tag{5}$$

From relations (4 - 5) it is clear that for the HMCVT with a non-reversible control characteristic, e.g. for the second traction range, if $\mathrm{sign}\frac{d}{dV}\left[e(V)\right]>0$ at the start of movement e_2 = -1, V_2 = 0, at e_2 = 0, $V_2 = V_{2\max}/2$, at e_2 = +1, $V_2 = V_{2\max}$.

Dependencies (4) and (5) are also functions of time t:

$$e_j\left(t\right)=\left(2\frac{V(t)}{V_{j\max}}-1\right)\cdot\mathrm{sign}\frac{d}{dV}\left[e_j(V(t))\right]; \tag{6}$$

$$V(t)=\frac{V_{j\max}}{2}\left[e_j(t)\cdot\mathrm{sign}\frac{d}{dV}\left[e_j(t)\right]+1\right], \tag{7}$$

where $V_{j\max}$ is the maximum speed of the machine in each speed range.

Thus, an algorithm is presented to search for the form of change in the hydraulic machine control parameter as part of the HMCVT.

4. Results

In On the basis of the formed mathematical model of the movement of a wheeled tractor during traction work, i.e. ploughing, it is possible to determine the change in its main operational indicators with various forms and intensity of the acceleration characteristic. Fig. 3 shows the change in acceleration of a wheeled tractor during the acceleration phase of the wheeled tractor. As can be seen from Fig. 3, the initial acceleration values have significant oscillatory effects accompanied by transient processes, so the maximum filtered values will be used in the remainder of this paper.

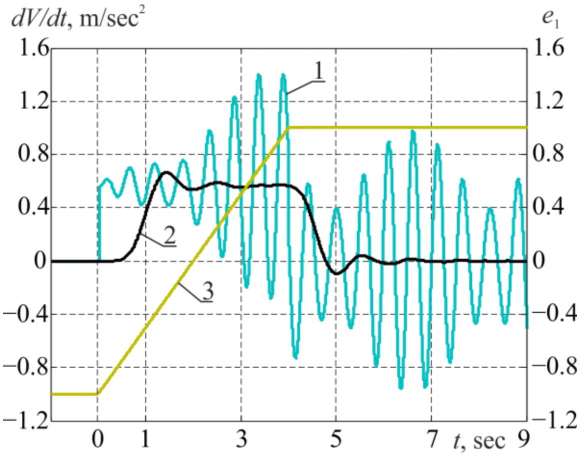
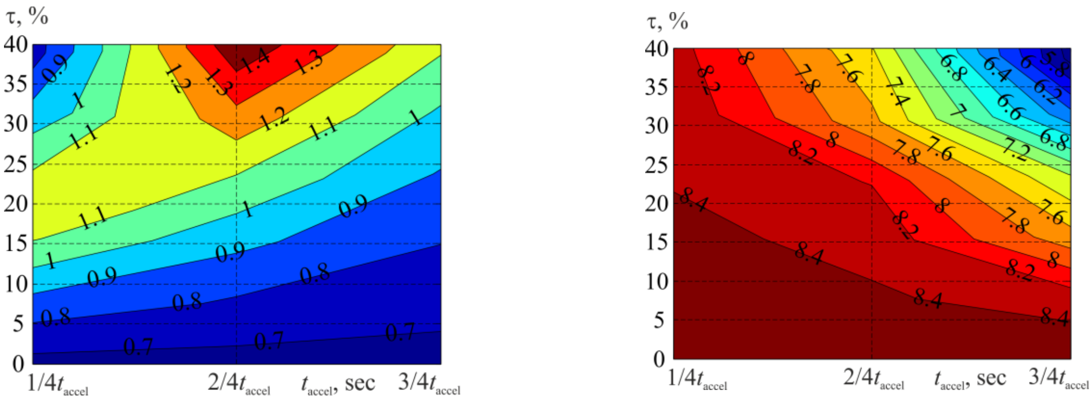


Figure 3 - Acceleration of a wheeled tractor in the acceleration phase: 1 - initial values; 2 - filtered values; 3 - change in hydraulic machine control parameter e_1 over time t_{accel}

Figures 4 - 6 show the changes in the maximum values of the operating indicators of a wheeled tractor when using different forms (Fig. 1) and the intensity of the acceleration characteristic curve τ. In particular, Fig. 4 shows the changes in the maximum values of the horizontal accelerations and the distance travelled during the acceleration phase.

Evaluating the results in Fig. 4 – 6, we note that there is no clear trend in choosing the convexity zone (¼t_{accel}, $\frac{2}{4}$ t_{accel}, ¾ t_{accel}).



a b

Figure 4: Dependence of the change in the maximum values of horizontal accelerations (a) and the distance travelled (b) during the acceleration phase, taking into account the use of different shapes (Fig. 1) and the intensity of the acceleration characteristic τ

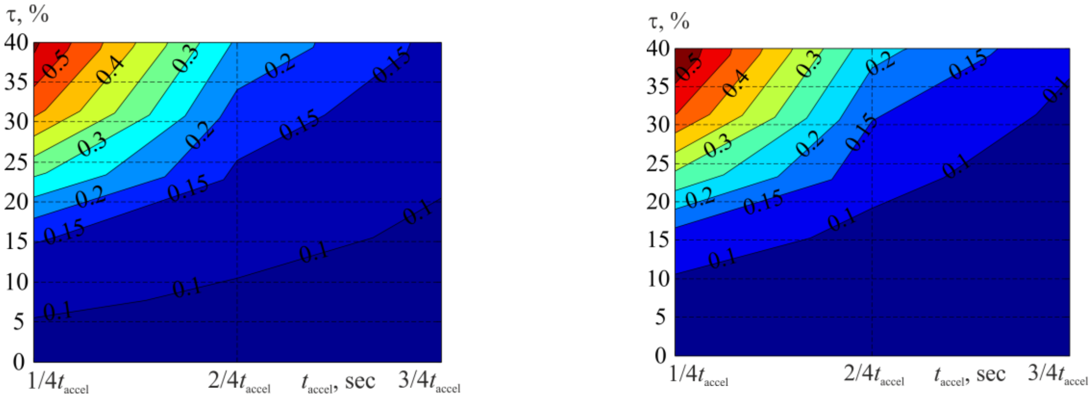


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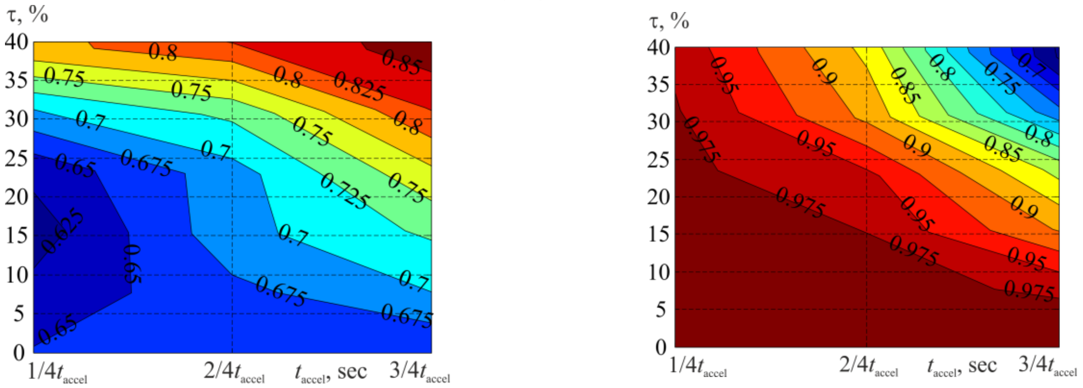
Figure 5: Variation of the maximum slip values of the front (a) and rear (b) tyres of a wheeled tractor during the acceleration phase, taking into account the use of different shapes (Fig. 1) and the intensity of the acceleration curve τ



a b

Figure 6: Variation of the maximum values of the efficiency of a wheeled tractor (a) and the relative acceleration time indicator $t_{accel}^{\tau=\mathrm{var}}/t_{accel}^{\tau=0}$ (b) taking into account the use of a different shape (Fig. 1) and the intensity of the acceleration characteristic τ

5. Discussion

Table 1 provides a qualitative assessment of the research results obtained, where "+" indicates an improvement in technical and operational indicators, "+/-" - mediocre values, and "-" - deterioration in technical and operational indicators.

Evaluating the convexity variants of the forms of modification of the hydraulic pump control parameters, we note that the improvement of the technical and operational indicators is observed with a convexity of ¾ t_{accel} of acceleration.

It was found that the use of a convex shape for ¾ t_{accel} of acceleration with intensity τ compared to the linear shape (τ = 0%) provides:

- increase in the maximum acceleration of 27.82% (at τ = 20%) and 68.4% (at τ = 40%);
- increase in the maximum slip value for the front wheels by 26.3% (at τ = 20%) and 70.9% (at τ = 40%) and for the rear wheels by 47% (at τ = 20%) and 115.7% (at τ = 40%);
- increase in efficiency of 15% (at τ = 20%) and 32.3% (at τ = 40%);
- reduction of acceleration distance by 13.3% (at τ = 20%) and 35.6% (at τ = 40%);
- reduction of acceleration time by 13% (at τ = 20%) and 35.5% (at τ = 40%).

Table 1 –Assessment of research results

Indicator	Convexity of form at ¼ t_{accel}		Convexity of form at $^2\!/_4$ t_{accel}		Convexity of form at ¾ t_{accel}	
	Expert assessment	Maximum values at τ = 40%	Expert assessment	Maximum values at τ = 40%	Expert assessment	Maximum values at τ = 40%
Acceleration dV/dt, m/sec^2	+	0,75	–	1,48	+/–	1,12
Wheel slipping δ_1	–	0,56	+/–	0,241	+	0,134
Wheel slipping δ_2	–	0,545	+/–	0,22	+	0,11
Efficiency η_{MTA}	–	0,79	+/–	0,826	+	0,863
The way of acceleration S, m	–	8,29	+/–	7,4	+	5,52
Relative acceleration time $t_{accel}^{\tau=\mathrm{var}} / t_{accel}^{\tau=0}$	–	0,968	+/–	0,867	+	0,645
CONCLUSION	Do not use		Can be used		Recommended for use	

6. Conclusions

A detailed method for working out the acceleration characteristics of a wheeled tractor with dual-flow non-reversible HMCVT transmission is presented. This is achieved by changing the hydraulic machine control parameter. It was determined that using a convex shape for ¾ t_{accel} of acceleration increases the technical and operational indicators of the wheeled tractor when performing traction work. It is also shown that the shape of the acceleration curve has an effect on the maximum values of tractor acceleration, wheel slipping (within safe limits) and efficiency. In addition, the acceleration path and time are reduced.

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